The

ENGINEERS' DIGEST

REVIEW OF WORLD ENGINEERING PROGRESS

VOL. XVIII No. 4 APRIL, 1957

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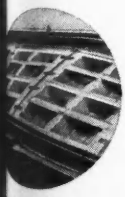
roved Performance with the 'L' Stoker



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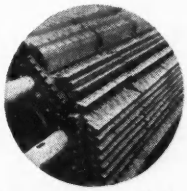

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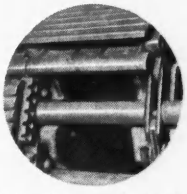

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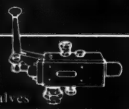

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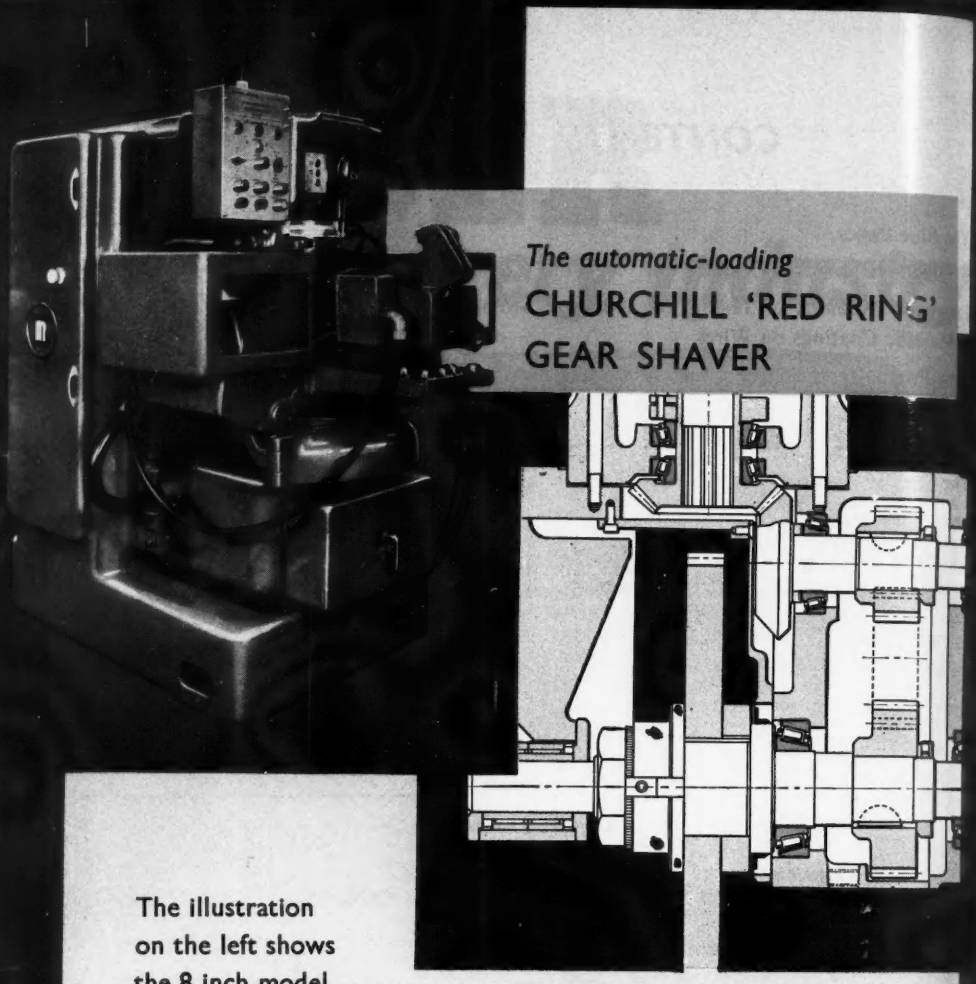

The automatic-loading

CHURCHILL 'RED RING' GEAR SHAVER

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THE BLUE PAGES

A QUICK GLANCE AT THE LATEST TECHNICAL DEVELOPMENTS THROUGHOUT THE WORLD

• *ANTI-WEAR COATINGS FOR ROLLING SURFACES*

Surfaces rolling over one another, such as the running surfaces of ball and roller bearings and the supporting contact surfaces of gear-wheels, are subjected to heavy stresses since, during rolling, only point or line contact occurs. This results in exceptionally high local loads, which greatly hinder the building up of a continuous film of lubricant. Through the unavoidable slip occurring during rolling, wear phenomena occur, in addition to destructive phenomena produced by dynamic stress. Special greases are therefore used for ball and roller bearings and gear-wheels, to ensure separation of the metal surfaces rolling against one another. It has now been found that especially effective protection against wear caused by slip can be obtained by providing the surfaces with a coating of nitride, boride, or silicide of titanium. The application of these coatings is not confined to steel surfaces, but other materials, e.g., bronzes, can similarly be treated. It is claimed that such a coating is extremely resistant to wear and possesses the necessary strength and internal cohesion to withstand any shock stresses which may occur. It is stated in the patent specification covering this process that a coating of this type can be applied to both the races and to the balls or rollers of a bearing, and that in this case a coating thickness of 200 to 400 micro-inches is sufficient. A coating can also be applied to the ball or roller cages, and for this purpose a coating of titanium nitride has been found particularly efficacious. The coatings are applied by a gas-phase process which is carried out at temperatures between 1000 and 1050°C in an electric furnace or by means of induction heating.

• *PRESSURE-CONTROL SWITCH WITH PULSATION FILTER*

A recent paper describes a pressure-control switch which uses viscous damping to prevent the transmission of pressure pulsations from a pump outlet system to the pressure-control system, this is being accomplished by designing a pressure switch with certain dynamic, as well as the normal static, characteristics. By controlling the amplitude of the pressure pulsations at the switch with a mechanical low-pass filter, a considerable simplification of the switch structure has been obtained. Several devices have been in use to reduce pressure pulsations, such as ball-type check-valves and small-orifice plugs, these devices aiding in reducing pressure peaks by an amount relative to the size of the opening through them. Since the size of the opening in these devices must be kept large enough to prevent clogging with foreign matter and to ensure proper operation, the resulting pulsation reduction is not very great. An interesting feature of the development work carried out is that an *RC* type of electric network was used as the basis of the pulsation-damper design. The two points of interest in the design are that the static pressure must be transmitted across the "filter" and pulsations of the frequency of the pump must not pass through the filter. The simplest form of an equivalent electrical filter is a resistance and capacitance in series. The resistance, mechanically, is a fluid of a particular viscosity flowing through an orifice of certain dimensions. The viscous damper uses a fluid contained between two resilient diaphragms, which, with their respective spring systems, form the capacitor part of the filter. The control switch developed, which was designed for a static-pressure range between 0 and 60 psi and a pressure differential between 13 and 40 psi, was found to be capable of reducing pressure pulsations by 90% or better, and the steady-state phase angle between the pump pulsations and the reduced pulsation at the switch was about 85 deg.

• *RADIOGRAPHIC DETECTION OF POROSITY IN PLATED COATINGS*

Pores and other flaws seriously impair the quality of industrially produced electroplated coatings, since they diminish their effectiveness as a protective barrier. Until now it has been difficult to estimate the relative merits of electrodeposits, as no method of determining the seriousness of defects was available. Recently, however, a non-destructive procedure has been developed which makes it possible to ascertain both the size and the position of pores present. This method involves photographing an electroplated specimen exposed to radiation. In determining the location and size of the pores, the specimen, which is a flat sheet of plated metal, a few thousandths of an inch thick, is placed on a photographic film with the coating against the film emulsion. The assembly is held in a cardboard cassette, and the basis-metal side of the specimen is exposed to soft X-rays or to radia-

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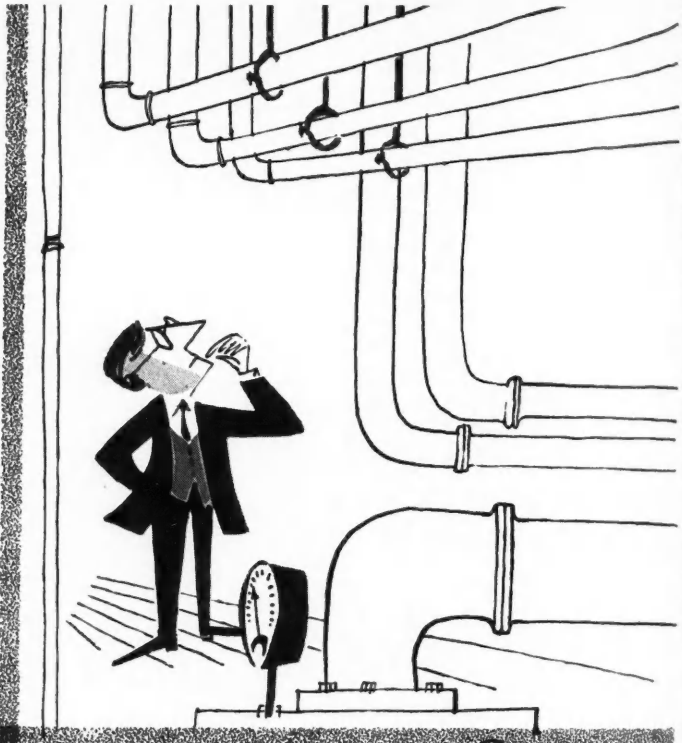

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tion from a radioactive material. This requirement for low-energy X-rays, to ensure good contrast, limits the thickness of the basis material to a few thousandths of an inch, because thicker metals absorb too much of the X-ray beam and require too long an exposure time. Radioactive iron, nickel, or cobalt held against the plated specimen in a cassette are also adequate sources of radiation. Satisfactory radiographs are claimed to result from exposures to X-rays of 3 to 4 min. Wherever a pore exists in the coating, a black spot appears on the developed film. It is claimed that this radiographic method detects pits, voids, and inclusions in the coating of the order of 0·001 in. in diameter. Discontinuities may also be determined by radiographing the basis metal before plating.

● *HYDROGEN EMBRITTLEMENT OF TITANIUM-MOLYBDENUM ALLOYS*

Hydrogen embrittles both unalloyed and alloyed titanium. In unalloyed titanium this is revealed by the impact test, the absorbed energy decreasing with increasing hydrogen content. In alloyed titanium, specifically in alpha-beta alloys, hydrogen embrittlement is revealed by slow tensile tests or creep-rupture tests ; with increasing hydrogen content the reduction in area decreases, sometimes quite sharply. Although the mechanism of hydrogen embrittlement of alpha-beta titanium alloys is not understood, a recent report states that a number of experiments has been made to elucidate the influence of specific alloy additions, total alloy content, test temperature, strain rate, and hydrogen content on hydrogen embrittlement. It is pointed out that all-beta alloys have a higher tolerance for hydrogen than alpha-beta alloys, and in the latter category the beta structure with molybdenum in solution maintains a considerably higher ductility than beta containing manganese. In alpha-beta titanium alloys, molybdenum also appears to promote greater tolerance for hydrogen than does manganese. Similarly, a 5% molybdenum or 11% molybdenum alloy seems to have greater tolerance for hydrogen than a 2% iron, 2% chromium, 2% molybdenum titanium alloy. Aluminium additions appear to increase the tolerance for hydrogen ; for example, a 6% manganese alloy is severely embrittled by about 270 ppm of hydrogen, whereas a 4% manganese, 4% aluminium alloy is not embrittled until 1000 ppm of hydrogen are introduced. A titanium-molybdenum alloy containing 30 to 40% molybdenum has been patented, and it is claimed that the 30% Mo alloy resists boiling 40% sulphuric acid and boiling 20% hydrochloric acid as effectively as platinum, tantalum, or gold. Such a retained beta alloy is weldable and could be fabricated into sheet.

● *METHOD OF ACETYLENE PRODUCTION BY TRANSIENT ELECTRIC ARCS*

A new method for the production of acetylene by means of transient electric arcs in mineral oil has been developed. The apparatus for this purpose is reported to consist of a hermetically sealed container of sheet metal, in which is mounted a frame, the bottom of which is a grid formed of carbon rods arranged in parallel. These carbon rods are connected to a source of alternating or direct current in such a way that adjacent rods have different polarity. The container is filled with mineral oil, and a number of balls of electrically conductive material are located upon the grid so that adjacent carbon rods are short-circuited. As contact is not perfect, small arcs are formed at the points of contact, with the result that oil is decomposed and gas bubbles are formed, lifting the balls from their seat on the rods and throwing them upwards, thereby extinguishing the arc. The entire process from the commencement of arcing to its termination is claimed to take place in less than ten milliseconds, so that the production of arcs takes place in rapid succession. The ensuing decomposition of the oil results in the production of some 29% acetylene, 50% hydrogen, and small amounts of ethylene, propylene, and butylene, as well as lampblack. By carrying out the process at sub-atmospheric pressure in the container, the acetylene content can be increased to 40%. Power consumption is given as 9·6 kWh per kilogram of acetylene.

● *ELEVATED-TEMPERATURE DRAWING*

Elevated-temperature drawing is the name of a new process which permits the production of steels with a unique combination of physical and mechanical properties. The properties obtained are stated to depend upon the careful selection of a particular composition of steel, the control of the amount of deformation carried out by reduction, and the temperature at which deformation is carried out to bring about metallurgical changes resulting in a new combination of physical and mechanical properties not heretofore available. It is claimed that, for the first time in steel-bar production, the dimensional tolerances, surface finish, and excellent machinability generally associated with cold-finished bars have been incorporated into a bar product which also has the strength properties normally developed by heat treatment, i.e., by quenching and tempering. Elevated-temperature drawing involves the selection of bar composition, the amount of reduction in cross-sectional area of the bar as it is drawn through a die, and a drawing temperature which will result in the desired final properties. These properties occur as a result of three metallurgical phenomena which take place in the bar as it is de-

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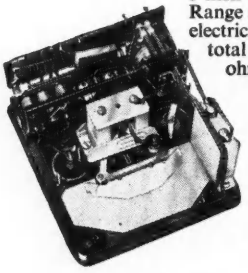

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formed at an elevated temperature between 200°F and below the lower critical temperature. The steel bar first strain-hardens by appropriate slip processes within the crystal structure ; it then age-hardens by precipitation of nitrides, together with precipitation of carbon from supersaturated solid solutions ; finally, a substructure within the grains develops as a result of the deformation at elevated temperatures.

● *HIGH-TEMPERATURE ELECTRIC FURNACE*

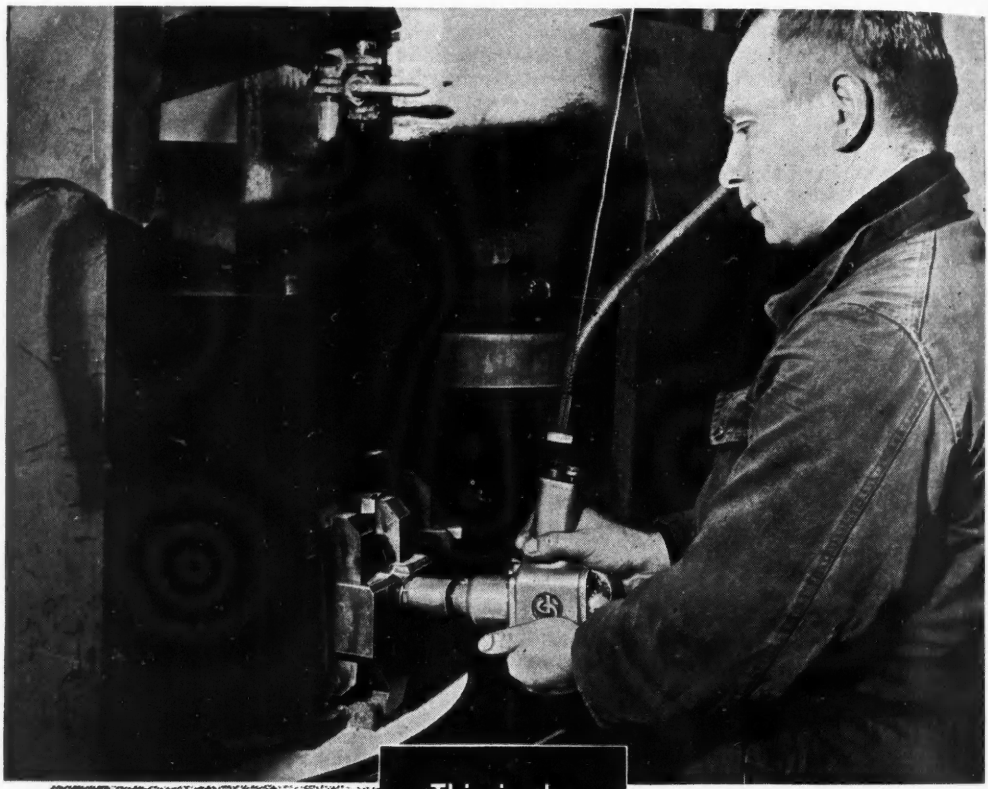
In high-temperature electric furnaces of the resistance type, the resistance wire used is susceptible to oxidation, so that its service life is reduced. This disadvantage can be circumvented by placing the windings in an inert or reducing atmosphere which usually also surrounds the material to be heated. In order to protect the windings, the provision in their vicinity of substances which readily combine with the atmospheric oxygen present has also been suggested. Another proposal is to apply to the wires surface coatings which are resistant to oxygen at the high temperatures prevailing. In a recent patent the point is made that a protective gas has the disadvantage that plant for the production and continuous supply of the gases must be provided. Also, measures must be taken to dry the gases and to remove admixed oxygen. The proposal is therefore made that the material to be heated be placed in a muffle in the furnace cavity and that the muffle be sealed against the infiltration of air. Heating of the muffle is accomplished by a heating coil of tungsten or molybdenum applied to the outer surfaces of the muffle. The walls of the furnace proper are constructed of highly refractory material and the furnace cavity is made gastight. This furnace cavity is filled with mercury vapour circulated through the furnace, so that the heating wires on the outside of the muffle and located in the furnace cavity are continually swept by mercury vapour. Circulation of the mercury vapour is accomplished by an external closed circuit drawing mercury vapour from the top of the furnace cavity and returning condensed, i.e., liquid, mercury to the bottom of the furnace cavity, where it is again evaporated.

● *FLAME HARDENING OF STEEL BY MEANS OF ROCKETS*

A recent report describes investigations carried out into the flame hardening of steel by the use of small rockets fuelled by oxygen and propane, the high-velocity, high-temperature jet from the rocket producing heat-transfer rates nearly four times greater than those of open oxyacetylene flames. This makes it highly important to ensure even distribution of the flame over the surface to be hardened, and for this purpose a slot-orifice rocket has been developed. Propane and oxygen are metered into the unit at pressures of 80 and 100 psig respectively. Slots over 1 in. long and 0·04 in. wide have proved successful, although the tests conducted were limited to a slot size of $\frac{5}{8}$ in. by 0·04 in., the jet temperature being approximately 5400°F and the jet velocity about 4000 fps. These figures represent a tenfold increase over the stream velocity of the oxyacetylene open flame. Higher jet velocities may be obtained by providing an expanding nozzle beyond the minimum throat section of the rocket. Uniform movement of the rocket over the metal surface is provided by a machine cutting-torch carriage restrained to linear travel, speeds between 20 and 60 ipm being possible. The difficulty of ensuring uniform jet distribution in linear hardening is absent in rotary hardening, owing to the rotation of the piece to be hardened, thereby giving every area of the surface an opportunity to be heated equally. It is believed that the high hardness values obtained with rocket hardening are due to the rapid heating cycle, as the rapid heat input allows the material to remain above the transformation temperature for a short time, with the result that little grain growth is experienced.

● *DRAWING-DIE SULPHIDE LUBRICANT*

It is well known that the plastic deformation of the metal in wire drawing is assisted by the presence of sulphide coatings. A recent invention envisages the formation of a sulphide coating as the wire is being drawn through the die by utilizing the heat generated during the deformation of the wire. For this purpose a highly reactive concentrated emulsion of colloidal sulphur in water is proposed. As, in the deformation process, a sulphide coating is also formed on the working surface of the drawing die, the latter may be made of low-cost steel instead of highly expensive tungsten carbide. The reactivity of the sulphidizing agent with the metal surface is to a high degree dependent upon the temperature, and the use of cold tools at the commencement of plastic deformation may therefore result in the formation of too thin a coating until the temperature has been raised by the heat of deformation. It is therefore proposed that the tool should be heated prior to commencement of deformation. On the other hand, the temperature at the working surface can then be regulated by varying the proportion of sulphidizing agent to water. Generally, water content should not exceed 10%, but some water must be present for the reaction to take place at all. Optimum water content is said to range from 1 to 3%. Friction of the work with the die and the hardening effect of cold working may be further decreased by using in addition a conventional lubricant. For cold working high-carbon or alloy steel a mixture of 60 to 90% by weight of colloidal sulphur and 40 to 10% of calcium stearate is recommended.



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●*PROPERTIES OF CORRUGATED DIAPHRAGMS*

Metallic diaphragms, singly or in pairs as capsules, find a wide use in pressure-measuring instruments of many types, their advantages including compactness, simplicity, cheapness, and a wide range of load, sensitivity, and deflection characteristics. The use of corrugated discs makes possible larger deflections and affords a control of the shape of the load-deflection curve. In practice, however, there are certain limitations regarding the use of diaphragms in any given application, because there are no generalized design formulae to predict the behaviour of corrugated diaphragms. This difficulty is serious, because it is always desirable to use the diaphragm best suited to a specified application, and the selection of the best form of diaphragm by trial and error may involve considerable development expense. There are two methods of determining the properties of corrugated diaphragms, i.e., the empirical and the analytical. The empirical approach is to make diaphragms of various sizes, shapes, and materials and to measure their properties. From judicious selection of sizes and shape for experiment, design criteria and generally applicable formulae to be used in the design of other diaphragms of similar shape and material can then be deduced. The analytical approach requires the derivation of equations relating the load-deflection characteristics to the geometry of the diaphragm and the elastic constants of the material. A recent research report gives the pressure-deflection characteristics of corrugated diaphragms as correlated by methods of dimensional analysis, making it possible to compute a diaphragm from a dimensionless formula derived from experimental data on other diaphragms of the same shape. These findings should prove highly useful to designers.

●*PROCESS FOR INCREASING RESISTANCE TO SCALING*

It is known that scale formed by the high-temperature oxidation of a metal surface provides only in very few cases effective protection against further corrosion by oxygen. Either the oxygen diffuses right through the layer and reacts further with the base metal, or the base metal travels outwardly in the form of ions and electrons and in this manner continues corroding. Research has shown that such a diffusion of ions and electrons through the crystal lattice of a layer of scale is possible only when various positions in the lattice are unoccupied or when additional ions can be incorporated between the actual lattice positions in interstitial positions. In electron-conductive layers, such as those formed by oxides, the partial conductivity of the ions is small, compared with that of the electrons. Thus, the rate of oxidation of metals capable of forming such layers is determined solely by the partial conductivity of the ions and therefore becomes small when a small lattice deficit concentration of ions is present. It has been found that the rate of scaling in an atmosphere containing oxygen can be reduced by introducing metal ions into the oxidized layer, so as to increase the electrical conductivity of the scale layer. It is claimed that, by means of this process, which is patented, it is also possible to increase the resistance to oxidation of titanium or alloys with a high titanium content. For this purpose a metal of higher valency than titanium, e.g., tungsten, is employed. The formation of the scale-resistant layer is carried out in air at 900 to 950°C by applying a powder consisting of aluminium oxide, titanium oxide, and tungsten trioxide. Thorium, cerium, beryllium, or calcium oxides can be added to improve the adhesion of the oxide layer. This process could be applied to gas-turbine blading made of titanium.

●*ELECTRONIC CLEARANCE INDICATOR FOR STEAM-TURBINE ROTORS*

An electronic instrument has been developed which continuously measures the clearance between the rotating and the stationary blades inside a steam turbine. This development was undertaken in order to assist in the study of steam turbines under actual operating conditions. One of the problems in steam-turbine operation is that of maintaining a safe axial clearance between rotating and stationary blades. The rotor- and stator-blade assemblies are supported respectively by the massive rotor and the relatively thin outer casing. Thus, transient thermal conditions can cause large differential expansions, resulting possibly in interference between the blades. The new electronic indicator permits accurate determination of blade clearance by measuring the axial distance from the shrouding around the rotor-blade tips to the base of the outer stator blades. Because of the severe conditions prevailing at the blade tips, caused by high-velocity steam at high temperature, use is made of a temperature-resistant sensing device of the mutual-inductance type, containing two coplanar, coaxial coils wound on a dielectric core. A radio-frequency source energizes the primary coil, the a.c. voltage induced in the secondary coil depending on the distance from the probe to an adjacent electrically conducting reference plate. The output voltage from the secondary coil is amplified and indicated on a meter calibrated in inches of clearance. To measure the transverse motion of the shroud band the primary winding induces voltages in two secondary coils which cover separate halves of the primary winding area. These half-windings are connected in series opposition, so that there will be no voltage output when the band around the blades is centred over the probe.

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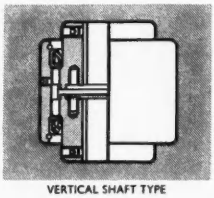

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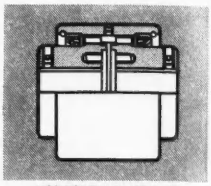

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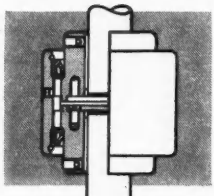

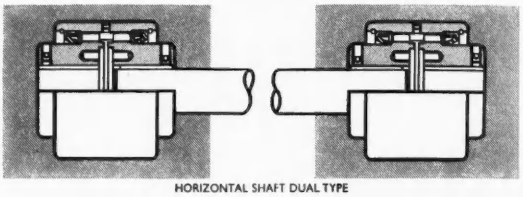

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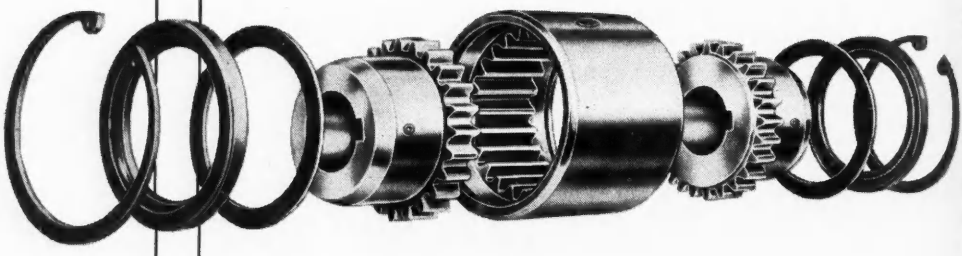

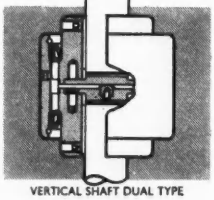

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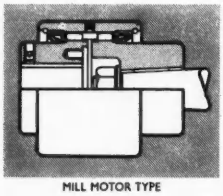

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THE ENGINEERS' DIGEST
REVIEW OF WORLD ENGINEERING PROGRESS
VOL. XVIII No. 4 APRIL, 1957

Magnetostriction as a Design Tool

By E. M. WISE. (From *Product Engineering, U.S.A.*, Vol. 28, No. 1, January, 1957, pp. 162–166, 6 illustrations.)

WHEN placed in a magnetic field, certain ferromagnetic materials undergo a change in dimension. This process, known as magnetostriction, is reversible, i.e., a change in magnetization occurs when the dimensions of a magnetostrictive material are varied by an external force.

The magnetostrictive device or transducer can be made in several forms, depending on the application. Of these, the simplest type is a rod of a highly magnetostrictive material such as nickel, excited by a solenoid winding. However, this form is limited to low-frequency applications, if high efficiency is sought. The laminated-stack type, though somewhat more complex in construction, has lower eddy-current losses and other advantages. In some instances, it may be possible further to simplify and improve a design by having a structural element serve as the core.

Compared with other methods of converting electrical to mechanical energy, the magnetostriction transducer has several advantages, as follows :—

(1) Ability to drive high-impedance loads, such as liquids and solids.

(2) Durable construction, similar to that of a transformer, so that it is not likely to be damaged by periodic overloads, loss of cooling, or neutron irradiation.

(3) Efficient operation, not only in a frequency range (5 to 100 kcs) where other transducers are relatively inefficient or difficult to construct, but also under difficult environmental conditions, such as high temperature, moist or corrosive atmospheres, dirt, water immersion, and large hydrostatic pressures.

(4) Adaptability to a wide range of sizes and power ratings.

(5) Flexibility of design, permitting the selection of different core shapes and the use of a variety of current and voltage levels.

Factors determining the magnitude of the force generated by a magnetostriction transducer, and the amplitude and frequency of the displacements include core material, service temperature, dynamic strain, losses, and configuration of the transducer.

Metals exhibiting magnetostrictive effects of practical proportions include nickel, cobalt, and certain alloys of these metals. Nickel exhibits one of the largest magnetostrictive length changes, about thirty parts in one million for a non-restrained body magnetized to saturation. Thus, a nickel rod one foot long would decrease in length by 0·00036 in. Constrained at both ends, this bar would develop a static stress of about 1000 psi.

Comparative data for the magnitude and sign of longitudinal changes in nickel, iron, and cobalt are given in Fig. 1. Iron expands in a low-strength magnetic field and contracts in a high-strength field, limiting its usefulness in many magnetostriction applications. Nickel contracts under all conditions, as does annealed cobalt, although the effect is small with cobalt. The behaviour of cast cobalt is somewhat questionable.

Considering all factors, pure nickel and the nickel-iron alloys are among the most useful of all magnetostrictive materials. Although iron shows both positive and negative changes in length L, the addition of a small percentage of nickel results in positive changes for all field strengths. More important, as shown in Fig. 2, the peak of the positive magnetostriction occurs at about 45% nickel. This peak value compares favourably with that of pure nickel, when this factor alone is considered. The 63% nickel alloy is the most strain-sensitive, i.e., it has the greatest change in magnetization for a given change in stress.

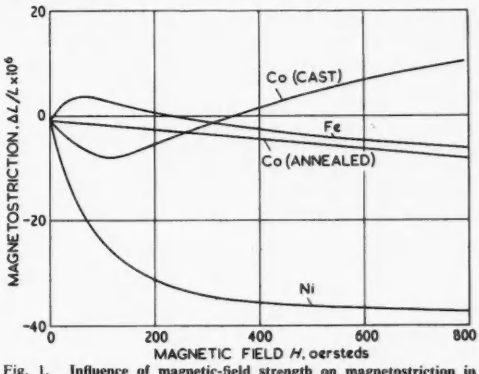


Fig. 1. **Influence of magnetic-field strength on magnetostriction in nickel, iron, and cobalt.**

With about 30% nickel (Fig. 2), longitudinal magnetostriction drops to zero (this composition is also non-magnetic at room temperature). With 81% nickel, the curve again passes through zero, and at higher nickel contents the changes in length are negative, reaching a peak value with pure nickel. This, combined with inherent corrosion resistance and substantial fatigue strength, has resulted in the use of pure nickel in most of the applications to date. Where fatigue strength must be further increased, Dura-nickel or one of the cobalt-iron alloys should be considered.

Volumetric changes have a different relationship to composition than the longitudinal changes referred to in Figs. 1 and 2. The change in volume is a maximum in alloys containing about 36% nickel. Invar, for example, derives its low thermal expansion from this effect ; loss in magnetization with increasing temperature alters the magnetostrictive effect and thus nearly neutralizes the thermal expansion.

Since the magnetic and magnetostrictive properties of a metal will be affected by changes in its internal struc-

ture, it may be necessary to specify heat-treatment procedures. In the case of nickel, a soft anneal generally gives the best magnetic properties, but the soft metal is difficult to punch for laminated construction. Quarter-hard material is preferred for punching, and the punched parts are annealed subsequently. An anneal may be specified to develop maximum magnetostrictive properties ; however, annealing reduces strength, and a balance may be necessary between acceptable magnetic and mechanical properties.

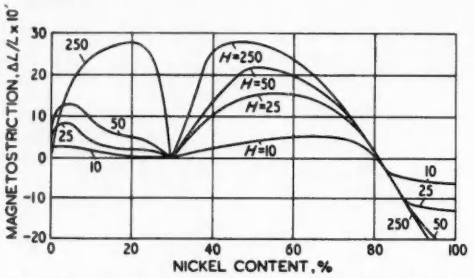


Fig. 2. Influence of nickel content on the magnetostrictive effects in a nickel-iron alloy system with different magnetic-field strengths H.

As the operating temperature increases, ferromagnetic alloys lose their magnetism, and thus their magnetostrictive properties. At the " Curie " temperature, these properties are zero for all practical purposes. Thus, for elevated-temperature service, a material with a high Curie point must be used. Data for the effect of temperature on the magnetostrictive effect for nickel are given in Fig. 3. Curves for other ferromagnetic metals have the same general shape, with 50 Ni-50 Fe remaining magnetic up to 520°C, and cobalt to about 1130°C. As a result, Co-Ni or Co-Fe alloys are generally used in elevated-temperature applications.

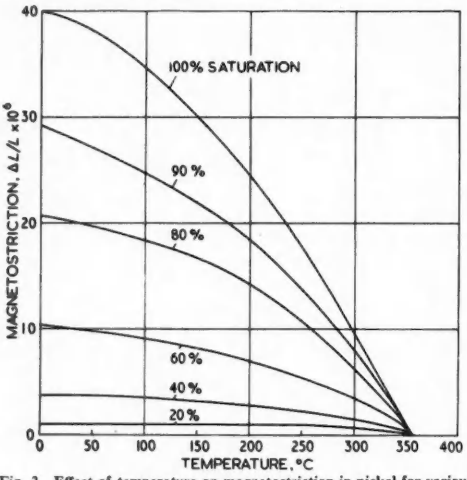


Fig. 3. Effect of temperature on magnetostriction in nickel for various degrees of magnetic saturation.

Mechanical and magnetic hysteresis losses and eddy-current losses often determine the permissible operating level of a magnetostrictive device. These losses will be determined to a large degree by the size and shape of the core, as well as the material. For maximum efficiency, the laminated type of construction is generally recommended. Eddy-current losses increase approximately as the square of the frequency and thickness, mixing the rod type inefficient at high frequencies. A thin tube will have lower losses than a rod, but power capacity is also lower. Stacked and insulated laminations about 0·010 in. thick are useful up to about 30 kcs ; approximately 0·003-in. laminations are preferred at higher frequencies. Above about 50 kcs, however, the thickness (as well as length) may be reduced so far as to make construction difficult.

Efficiency may vary from 90% for a well-designed power-type unit to 30 to 65%. Where maximum efficiency is required, magnetic bias should be provided. For example, the curves in Fig. 4 compare the performance of a typical low-power unit with and without bias. This bias can be supplied by a permanent magnet, by a separate d.c. supply, or by the d.c. component in the plate circuit of the driving amplifier.

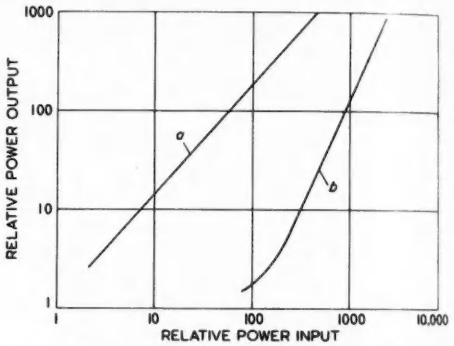


Fig. 4. Comparative performance of a typical low-power unit, (*a*) with bias, excited at resonant frequency, and (*b*) without bias, excited at half resonant frequency.

Permissible magnitude and frequency of displacements for a given material depend on its fatigue strength. However, the maximum operating stress must be considerably lower than the nominal endurance limit, since the useful life of a transducer involves many more cycles than are normally encountered in mechanical design. Whereas 10^7 or 10^8 cycles are generally considered as infinite life for a mechanical component, 10 hours of transducer service at 25 kcs would involve almost 10^9 cycles. In addition, a transducer core is not an ideal structure from the standpoint of fatigue. Sharp edges, changes in section, and welded joints between dissimilar metals create stress raisers. A practical operating stress for an annealed laminated nickel transducer core is 10,000 psi. This corresponds to a maximum strain of about 350 micro-inches per inch, or higher.

Operating frequency is as important as transducer geometry. Although a unit may operate over a range of frequencies, it is generally most practical to design for resonance operation. Under ideal conditions, the only energy required for resonant operation is for offsetting the losses and supplying the load on the transducer. The mechanical power generated in a transducer is directly proportional to the dynamic strain level.

Standard commercial equipment can be used to supply the current for generating the magnetic field. The actual field strength required will depend upon the material used. For nickel, 100 oersteds is a nominal value, although it is possible to operate at 50 to 200 oersteds for nickel, as well as for some of the more commonly used materials.

Practical Applications

The practical applications of magnetostriction were stimulated by World War II, sonar and related devices for detecting submarines and ships by pulse generation and reflection being perhaps the best known of such applications.

Although industrial applications of magnetostriction are a comparatively recent development, several significant examples are already in use. Perhaps the best known of these are in the field of cleaning and degreasing of metals.

Another significant development is the welding of similar or dissimilar metals by vibratory energy produced by magnetostriction. Pieces to be joined are clamped between two elements, and intense vibratory energy is introduced from a transducer ; the result is a solid-state metallurgical bond, with low deformation and no fusion. This method is particularly useful in joining very thin sheets to heavy sections—a difficult task with fusion welding.

In the soldering field, magnetostriction makes it possible to solder without flux, thereby reducing the possibilities of corrosion an l contamination. A magnetostrictive transducer, delivering high-frequency vibratory energy, is designed into a soldering gun equipped with a heated tip. This energy is transmitted to the soldering tip, causing cavitation in the molten solder which mechanically breaks down the oxide film and permits the solder to wet the base metal.

An equally interesting and important application involves the use of magnetostriction in a machine-tool feed device.* A solenoid winding is wrapped around a bar of nickel several inches long. Clamping the bar at either end alternately, and turning the current on and off results in an inching motion which is used to feed a workpiece into a grinding wheel. Increments of the order of a few millionths of an inch are possible.

An oil-well drilling unit now under development utilizes a nickel core weighing over 1 ton, demonstrating that size is not a limitation in application of magnetostrictive principles. This device utilizes the vertical vibration of a heavy bit to crush rock in its path, the particles then being flushed away with drilling mud.

A new application under development is an automotive fuel-injection system in which magnetostriction is used to produce pressure pulses which inject fuel into the cylinders at the right instant for firing.

* See also page 177.

GERMANY

Improved Methods of Testing the Adhesive Strength of Metallic Coatings on Wires

By E. A. Franke. (From *Metalloberfläche*, Vol. 11, No. 2, February 1957, pp. 53–59, 13 illustrations.)

To protect against corrosion, metallic coatings must be capable of adhering to wires throughout their life under all possible service conditions, without flaking or forming pores. The usual method of testing the adhesive strength of these coatings is to wind a length of the coated wire around a mandrel of about 8 to 10 times the diameter of the wire, or even less, and to ascertain whether the coating flakes off or is otherwise damaged. However, this type of test does not guarantee a durable adhesive strength under, for instance, fatigue or impact conditions, and, as a result, a number of suggestions for additional, more stringent tests are made in the following.

Torsion Test

The wire is twisted in a conventional torsion testing machine, one of the chucks being easily movable axially, so that no tension is imposed. The number of complete revolutions obtained before the first particles of coating loosen or flake off is a measure of the adhesive strength in torsion. For uniformity of results, a constant test length and rate of twisting are required, both chosen in accordance with the diameter of the wire under test.

Rolling Test

In the rolling test (Fig. 1), the test wire *c* is rolled between two cylindrical rolls *a* and *b* with a variable centre distance, so that the rolling pressure can be adjusted. One roll is driven manually or preferably by an electric motor at a constant slow speed, as a high speed would produce undesirable additional stresses and would cause premature flaking of even sound coatings. Since adhesion may not be uniform over the entire length of the wire, w ich feeds through the rolls by friction, the length of the test wire should be fairly long. This test should be carried out at intermittently increased pressures or with repeated passes at increasing pressure. The pressure at which the first particles of coating flake off completely can be taken as a measure of adhesive strength.

Bending Test under Tension

In this test, the wire *c* is held between two clamps *b* and is bent by pressure applied via a grooved plate or disc *a* (Fig. 2), guided transversely to the axis of the wire. The groove should be of slightly larger radius than the wire, so that it is preferable to use discs with different groove radii for various thicknesses of wire to be tested. The disc is pressed against the wire until the first particles of the coating flake off on the more highly stressed tension side of the wire, the deflection obtained constituting a measure of the adhesive strength of the coating. In view of the fact that even the best coatings will eventually flake off under this test, a certain minimum deflection may be stipulated as a required acceptance standard.

Improved Winding Tests

By using a conical drum *a* (Fig. 3), it is possible to

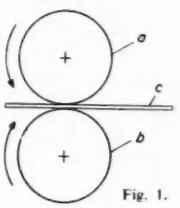


Fig. 1.

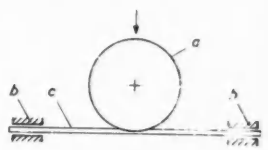


Fig. 2.

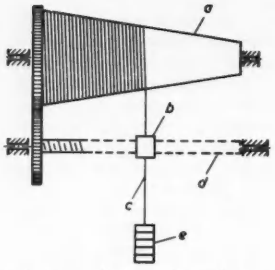


Fig. 3.

replace a whole series of the cylindrical mandrels of different diameters used in winding tests. The wire *c* is fixed to the wider end of the drum and is loaded by weights *e* at the free end, so that it is forced to maintain contact with the drum during winding and to straighten out during unwinding. On smooth drums, the wire must be guided by hand or automatically, as shown in Fig. 3, by means of a slide *b* travelling along a spindle *d* during winding and unwinding. On grooved drums, no guiding device is necessary. If no flaking is observed after winding, the wire may be unwound by reversal of the direction of drum rotation, and then rewound. The number of winding and unwinding operations at which the first signs of flaking occur is a measure of the adhesive

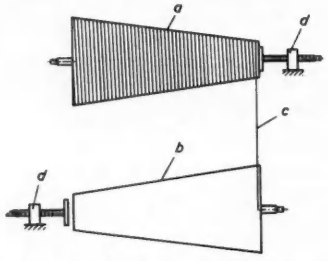


Fig. 4.

strength of the coating, and the diameter of the drum at the point where flaking is first observed must be indicated as a measure of the stress imposed. Suitable dimensions of the drum are 3 ft in length, and maximum and minimum diameters of 16 in. and $\frac{1}{2}$ in. respectively.

The winding test can be simplified and accelerated if two drums *a* and *b* are used, as shown in Fig. 4. In this method, the wire *c* is first wound onto one drum, after which the free end is attached to the other drum, which is then rotated so that the wire winds onto the second drum and off the first drum. To ensure that the wire is tensioned sufficiently for close winding, friction brakes *d* are provided, restraining the unwinding drum during rotation of the winding drum. Each complete winding cycle stresses the wire twice and, in general, stressing is unequal, as the same length of wire is wound onto different diameters. However, stressing can be made equal, if the drums *a* and *b* are positioned as shown in Fig. 5 and made to rotate at equal speeds by means of a simple gear or chain transmission, ensuring a constant tension of the wire *c*. As before, the number of complete cycles and the winding diameters at which flaking first occurs can serve as parameters for the test. Considerably increased stressing can be obtained if the drums are so arranged that the tension side of the wire on one drum corresponds to the compression side on the other drum. The stress then alternates between tension and compression values which are unequal (Fig. 4) or equal (Fig. 5).

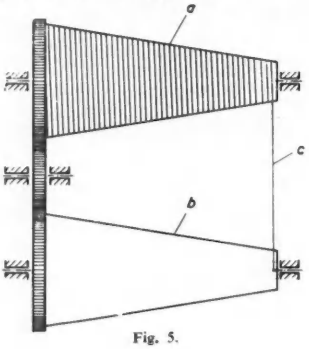


Fig. 5.

Impact Tests

If the wire is likely to be exposed to impact stresses, the adhesive strength of the coating should be tested for its endurance under this type of loading. Suitable test equipment may comprise a simple pendulum-type hammer or a hammer operated by a ratchet device *b* (Fig. 6), and in each case the wire *c* may be stretched flat on a plate or, as shown in Fig. 6, wrapped around a disc *d*. The plate or disc should be fitted with a groove to hold the wire, and the wire itself must be suitably clamped in position, so that it is mounted firmly under a certain initial tension. With this equipment, two methods of testing are possible, in one of which the impact of the hammer *a* is always directed against the same point on the wire until flaking is initiated, the number of impacts required being a measure of the adhesive strength. As this test must be repeated at several points along the wire, to obtain reasonably accurate results, these points must be adequately spaced apart (e.g., 8 in. on a wire of $\frac{3}{16}$ in. diameter), so that the results of the subsequent test are not influenced by the effects of the preceding test. In the second method, each point is impacted only once, the subsequent impact being directed against a closely adjacent point. The test wire is thus moved continuously under impact from one end of its length to the other, and the number of impacts required to produce the first signs of flaking is counted. The advantage of this method is that it provides uniform testing of the whole length of wire.

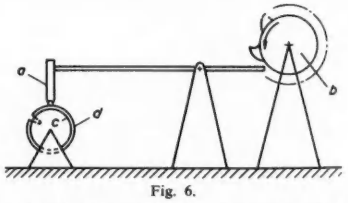


Fig. 6.

The ratchet-driven hammer is easily adaptable to automatic operation by means of an electric motor and revolution counter, but the impacts it produces are smaller than those obtained with a pendulum-type hammer, so that the number of impacts required to produce flaking is considerably higher with the ratchet-driven hammer. Once the mean strength of a known perfect coating has been determined, it is a simple matter to stipulate a minimum number of impacts which must be withstood without any signs of flaking or loosening. All that is then necessary is to run the hammer to this number of impacts and subsequently to examine the testpiece.

New Principles of Setting Up and Resetting in Machining Operations

By B. S. BALAKSHIN. (From *Vestnik Mashinostroyeniya*, No. 1, January 1957, pp. 44–49, 8 illustrations.)

IN machining operations, components form part of the kinematic and dimensional chains in the complete system, consisting of the machine tool, the fixture, the cutting tool, and the component itself. The complete system is appropriately set up when a new component is to be machined (setting-up operation); during machining, the original accuracy is progressively lost, and often the system has to be readjusted (resetting operation).

Each dimension of a component is fully specified when the tolerance δ is given, together with the coordinate Δ_0 which determines the position of the tolerance band in relation to the nominal size. These two quantities are generally expressed by the two limiting deviations *BA* and *HA*, where *A* is the nominal size (Fig. 1).

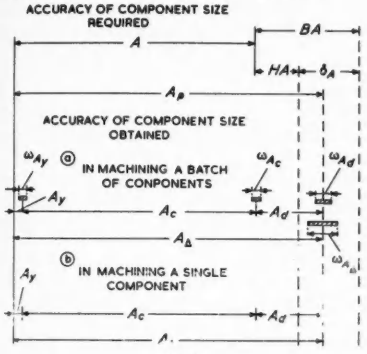


Fig. 1.

As a result of machining a batch of components after a single setting-up procedure, a scatter band ω_n (Fig. 2) is obtained, which may occupy some position in relation to the nominal size of the component, this band depending on the combined effect of a number of systematic and random errors. Suppose the systematic factor to be the dimensional wear of the cutting tool and the measuring instrument. The simplified diagram in Fig. 2 illustrates the fact that the combined effect of all the other factors creates a scatter band ω_r' which, under the influence of the dimensional wear of the cutting tool (assumed to be proportional to the number of machined components), gradually shifts across the tolerance band until it reaches the upper limit. At this point, resetting of the complete system (machine tool, fixture, cutting tool, and component) is required, to avoid scrap. Thus, the tolerance δ_A of dimension *A* (Fig. 2) must be divided into one part to cover the errors due to systematic factors, and into another part to cover random errors.

For setting up, it is customary to compute the so-called "working setting-up dimension" A_p, i.e., the dimension aimed at in setting up. It follows that the smaller is the magnitude of the assumed scatter band ω_r, the larger will be the number of components which can be machined with the same tolerance before resetting first becomes necessary. Thus, the task of setting up may be divided into two operations, i.e., (1) to obtain the minimum width of scatter band produced by random factors only, i.e., the width of the actual scatter band must not exceed the assumed width; and (2) the positioning of this final scatter band ω_r' in the desired relation to the tolerance band, i.e., the coincidence of the centre of the final scatter band ω_r' with the limit of the working setting-up dimension *A* (Fig. 2).

In practice, division of the setting-up process into three parts is necessary, i.e., (1) stationary setting up (without working loads) of the corresponding kinematic

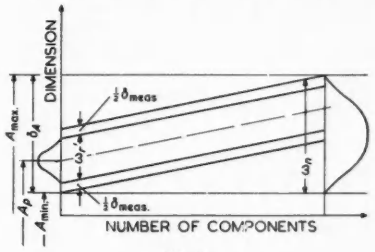


Fig. 2.

and dimensional chains of the complete system; (2) loading and clamping of each component in the machine tool or fixture; and (3) setting up under load (and other factors affecting dimensional relations) of the same points previously set up without load. Frequently, the first and second parts are reversed in sequence.

In the general case, each of the three elements is associated with its own nominal dimensions (A_c, A_y, and A_d) and tolerances (δ_{A_c}, δ_{A_y}, and δ_{A_d}). As a result of machining a batch of components, each element contains deviations in the form of scatter bands ω_{A_c}, ω_{A_y}, and ω_{A_d}, located in a certain manner about the nominal dimension.

Assume for simplicity that each scatter band is symmetrically disposed with relation to the nominal size of each element, as shown diagrammatically in the example of producing a size *A* in a batch of components (Fig. 1a), and in a single component (Fig. 1b). It follows from Fig. 1 that the mean size A_Δ of the component, obtained as a result of machining, is the closing link in the dimensional chain

$$A_c + A_y + A_d - A_\Delta = 0$$

so that the following two equalities can be stated:

$$A_\Delta = A_c + A_y + A_d \quad .. \quad .. \quad .. \quad .. \quad (1)$$

$$\omega_{A_\Delta} = \omega_{A_c} + \omega_{A_y} + \omega_{A_d} \quad .. \quad .. \quad .. \quad (2)$$

From eq. (1), it follows that, with a given setting-up dimension A_p and with known magnitudes of A_y and A_d, the value of the nominal stationary setting-up dimension A_c can be expressed as

$$A_c = A_\Delta - A_d - A_y = A_p - A_d - A_y \quad .. \quad (3)$$

If it were not for deviations, by setting up the cutting edge of the tool at a distance A_c from the base of the machine tool or fixture, which determines the position of the machined components, it would be possible to obtain immediately an accurate size of machined component. However, the existence of deviations demands the provision of tolerances for each of the nominal dimensions A_c, A_y, and A_d, and the actual deviations must be maintained within these tolerances in order to ensure the required accuracy.

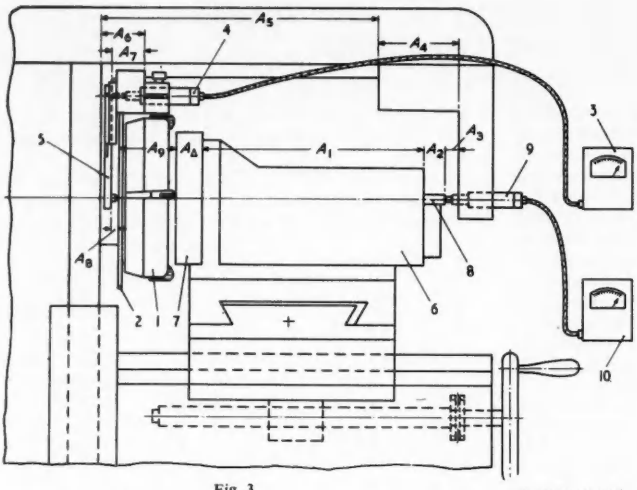


Fig. 3.

Using eq. (2), the tolerance for each link in the dimensional chain in eq. (1) can be established, starting from the width of the scatter band ω_r assumed in setting up, and substituting its value in eq. (2) :

$$\omega_r \leqslant \delta_{A_c} + \delta_{A_d} + \delta_{A_y} \quad .. \quad .. \quad .. \quad (4)$$

Numerous methods exist for measuring with satisfactory accuracy clamping and stationary setting-up errors in the dimensional and kinematic chains. Hence, there is no special difficulty in maintaining these deviations within their limits. Setting up, however, remains a difficult operation without the provision of means for accurate measurement of setting-up deviations from the dimension under load A_d. The amount of possible deviation from A_d is at present judged solely on the basis of tests. For this reason, setting up for the required accuracy of this dimension is approximate and depends entirely on the skill of the operator. The method of trial machining used in setting up consists of machining the whole or part of the surface of the component, measuring the dimension so obtained, and introducing the necessary corrections into the setting-up procedure, unless, of course, the result is already within the prescribed tolerance ; otherwise, the procedure is repeated until the resulting size is satisfactory from the point of view of tolerance. When machining a single component, setting up is simpler because, in this case, all that is required is to ensure that the deviation does not exceed the total tolerance δ_A.

The problem is considerably more complex when setting up the complete system for machining a batch of components, when the deviations of each component must remain within the limits of the closer setting-up tolerance, which forms only one part of the total tolerance δ_A. In this case, it is usually necessary to machine several trial components, so as to judge from the mean value of their dimensions whether the final scatter band ω_r' is correctly positioned in relation to the working setting-up dimension. This time-consuming process is not only inherently wasteful but also makes the introduction of high-production equipment more difficult in the manufacture of small batches of components.

In order to eliminate these drawbacks, a new system of setting up is proposed, as follows :

An " equivalent " force P_e produces, as a result of the elasticity of the complete system, a displacement A_d' at the cutting edges of the tool relative to the machined component. This displacement is a function of the equivalent force P_e, which, in turn, depends on cutting conditions and on the properties of the material. Thus, in order to control the magnitude of A_d' by displacing the tool, the most appropriate method is direct measurement of the distance between the cutting edge of the tool and the base of the machine tool or fixture. However, with the majority of existing machine tools this is as yet impossible. Nevertheless, the problem may be solved by using one of two methods of measuring the magnitude of the displacement A_d', i.e., (1) measurement of the relative displacements (or the force associated with them) between the components forming part of the appropriate dimensional or kinematic chain of the complete system; and (2) measurement of the elastic deformation of a special link (or links) having a known flexibility, inserted into a chosen dimensional or kinematic chain of the system.

As an example of the first method, Fig. 3 shows diagrammatically part of a universal horizontal milling machine fitted with experimental equipment for measuring the relative displacements between the milling cutter and the fixture in which the machined component is clamped, under the action of the equivalent force P_e arising during the cutting process. For this purpose, the cutter (1) carries a disc (2) turned *in situ*, i.e., after setting up the milling cutter in the machine. The axial displacements of this disc and consequently those of the cutting edge of the milling cutter are measured by means of a suitable instrument (3), whose pick-up element (4) presses against one of the arms of a lever (5), while its other arm touches the surface of the disc (2). A gauge-bar (8) is attached to the fixture (6) in which the component (7) is clamped, the bar being aligned in parallel with the motion of the work-table of the machine. The inductive pick-up (9) of the second measuring instrument (10), attached by means of a cruciform member to the yoke of the milling machine, incorporates a feeler which touches the surface of the bar. During the machining process, the displacement between the component in its fixture and the bed of the milling machine is constantly varying. The algebraic sum of the readings of both instruments gives the magnitude of the dsiplacement A_d' between the cutting edges of the milling cutter and the base of the fixture, determining the position of the machined component.

To illustrate the second method, Fig. 4 shows a special elastic link in the form of a flat plate (1) inserted into an experimental installation in the dimensional chain of a cylindrical grinding machine. The equivalent force P_e acting on the grinding wheel is transmitted to the grinding-wheel carriage (3) butting against a special stop (4) through the grinding-wheel spindle and its supports, the housing of the grinding-wheel spindlestock, the rack (2), and the pinion, shaft, and wormwheel transmission. The machine bed (5) carries two clamps (6), which serve as stops for the flat plate (1). The load is transmitted through the stop (4) to this plate and through its supports to the machine bed (5). In the

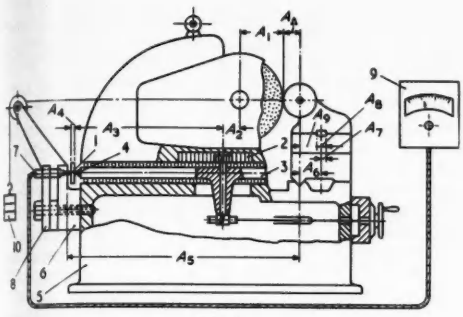


Fig. 4.

centre of the plate, an inductive pick-up (7), attached to a special clamp (8) fixed to the machine bed, presses against the plate, the pick-up being connected to the indicating instrument (9). A preload is introduced into the system by means of a weight (10) at the end of a wire rope passing over a pulley and connected to the housing of the grinding wheel, in order to stabilize indicator readings. After calibration, this equipment makes it possible to measure the force on the plate from the amount of its displacement and hence to find the magnitude of displacement of the grinding wheel in relation to the machine centres. The displacement is a function of this force, and determines the actual size of the machined components, i.e., the magnitude of A_d'.

Tests have established that both these methods can solve the problem outlined here with an accuracy which is adequate in practice. To illustrate the procedure adopted, the nomogram reproduced in Fig. 5 is based on investigations carried out in the machining of components on a cylindrical grinding machine. It can be seen that, by connecting cutting conditions, dimensional accuracy obtained, and properties of the machined material, it is possible to select grinding conditions which combine the desired accuracy with the maximum output possible for a particular component, machine tool, and grinding wheel. In fact, the first passes can be conducted, for example, at the highest output, limited only by the danger of overheating the ground surface.

It is easy to establish grinding conditions from the nomogram, starting from a grinding-wheel pressure which will not cause surface overheating. For instance, if the permitted magnitude of grinding-wheel line pressure is 5 kg/cm, the grinding-wheel pressure must not exceed 15 kg with a wheel of 30 mm width. The nearest line on the nomogram, i.e., 14·8 kg, corresponds to an instrument reading M of 40 microns. Assuming a depth of cut t of 24 microns, the corresponding value is found on the vertical scale, and a horizontal line is then drawn to its intersection with the sloping line defining the properties of the material (in this example, a hardened 0·5% carbon steel). From the point of intersection, a vertical line is dropped to meet the curve of permissible grinding-wheel pressure ($P_e = 14{\cdot}8$ kg). Proceeding along this curve until it intersects the longitudinal rate of feed s (e.g., 8 mm/rev), and then dropping from this point of intersection a vertical line to the lower scale, the surface speed v_w of the machined component, i.e., 23·5 m/min, is found.

Having thus established the grinding conditions, the component can be machined, observing the indicator reading M, which should be maintained at 40 microns. If the reading changes, it can be restored by varying, for example, the rate of longitudinal feed s, ensuring machining of the component with a constant grinding-wheel pressure and avoiding overheating. After rough-machining, the component is measured, to determine exactly the machining allowance which has to be removed in the finishing operation. When the number of finishing passes has been established, it is possible to find, with the aid of the same nomogram (at given values of the speed v_w and depth of cut t), the magnitude of the longitudinal rate of feed s, the required pressure P_e, and the corresponding reading M of the indicator. The solution of this problem is also shown (by the dotted lines) on the nomogram in Fig. 5.

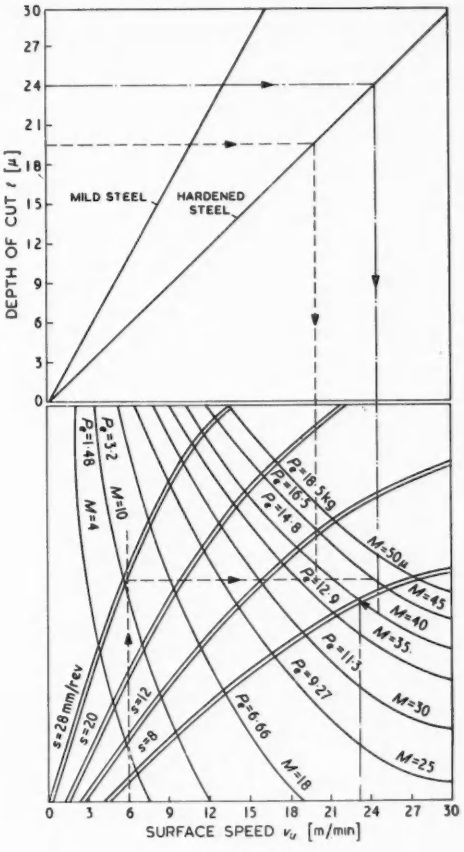


Fig. 5.

Additional scales can be incorporated in the nomogram, by means of which it is possible to determine cutting conditions, starting from the required surface finish and accuracy of the geometric form of the machined surface of the component. The finishing passes may be so chosen as to obtain optimum surface finish.

The investigations carried out in this connection have revealed a further potential reserve for improvement in machining accuracy at the same output. This reserve is based in principle on the elimination of a number of machining factors from the category of random effects, and on their transfer into the class of controllable systematic effects. These factors include variations in the

amount of machining allowance and the differences in the properties of the material of the separate components to be machined. At present, these values are counted, among others, as random factors in the machining of a batch of components, and so enter into the magnitude of the scatter band ω_r' of the component dimension.

In order to control the effect of these factors, it is necessary to measure the actual machining allowance and the hardness of every component before machining. It is then possible, from the results of these measurements, to introduce corrections into the cutting conditions, so that cutting can be effected at a constant value of the setting-up dimension under load, and in this way to narrow the scatter band. These relations are shown diagrammatically in Figs. 6 and 7, which apply to the machining of a number of components, A_b being the size of the blank before machining, and δ_b the tolerance. Fig. 6 shows the resulting accuracy of the dimension A obtained by the ordinary method, and Fig. 7 the accuracy of the same dimension obtained by the method of introducing

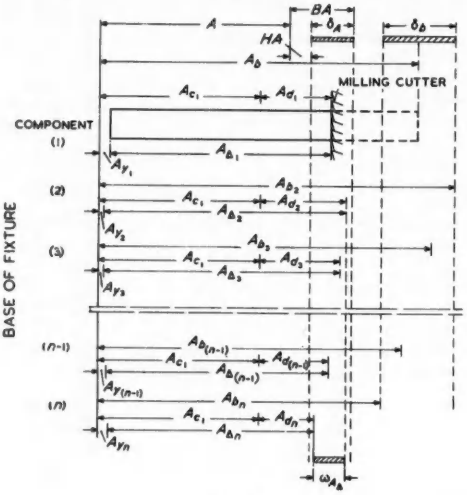


Fig. 6.

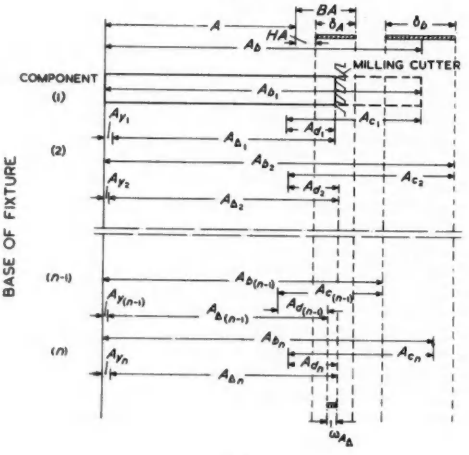


Fig. 7.

the necessary corrections into the cutting conditions derived from individual measurements of machining allowance and hardness, before starting machining. It can be seen from these diagrams that, in the first case, machining is carried out with a more or less constant value of the stationary setting-up dimension A_c. In the second case, not only the stationary setting-up dimension A_c, but also the setting-up dimension under load A_d, is constant, as, too, is the equivalent force P_e. As can be seen from these diagrams, a narrower scatter band is obtained.

In a similar manner, by measuring the temperature of the links in the dimensional chain (choosing a link with substantial temperature variations), it is possible to determine the magnitude of the deviations from the temperature assumed in setting up and, by introducing the necessary corrections into the cutting conditions, to compensate for the additional changes in the distance between the cutting edge of the tool and the machined component, these changes being associated with temperature fluctuations. The corrections can be introduced either manually or automatically.

It follows from this discussion that the insertion of individual measuring instruments into the dimensional and kinematic chain of the complete system makes it possible to introduce corrections in setting up which compensate for the effects of a number of originally random factors on the accuracy of machined components. In other words, this built-in instrumentation constitutes a manual or automatic resetting system and so maintains over a long period the necessary machining accuracy.

Together with the factors so far considered, dimensional wear of the cutting tool also affects the magnitude of the setting-up dimension under load during the process of machining a batch of components. Compensation for dimensional wear of the cutting tool is effected in various ways, all based on measurement of the component already machined and the introduction of corrections derived from this measurement. Usually, these corrections concern the depth of cut t, although other corrections are also possible, e.g., to the rate of feed s.

From the above considerations, it can be seen that resetting in automatic production lines consisting of a number of separate machine tools can be accomplished by means of the following system : Before commencing machining, the component blanks should be subjected to actual measurements of machining allowances and hardness. On the basis of these measurements, corrections are then introduced into the cutting conditions established when setting up, ensuring the specified accuracy at the optimum output possible with the equipment used. Components machined in the first machine tool are subjected to measurements of the actual magnitude of the machined dimension and, if necessary, hardness. Depending on the actual values so measured, the measuring system can then produce (1) a signal to the preceding machine tool for compensation of dimensional wear of the cutting tool, and (2) a signal to the subsequent machine tool for introduction of the necessary setting-up corrections required for improvement of the machining accuracy of the component advancing from the preceding machine tool. Such an installation must be arranged not only between each machine tool, but also ahead of the first and behind the last machine tool.

The principles discussed in this article are designed to ensure with given equipment the highest possible output consistent with a specified accuracy of machining operations.

Impulse Testing of E.H.V. Transformers at 1775 kV

By B. GÄNGER. (From *The Brown Boveri Review*, Vol. 43, No. 12, December 1956, pp. 525–532, 10 illustrations.)

It is frequently specified nowadays that a transformer must possess adequate impulse strength, the measures taken to make best use of the insulation under the stresses imposed by voltage surges determining, to a large extent, the design and arrangement of the windings and their connecting leads. This is particularly true in the case of transformers for extra-high voltages, where an entirely new line of approach has to be adopted, because conventional methods of control to obtain uniform stress distribution, though suitable for transformers up to 220 kV, are no longer sufficiently effective at higher voltages.

This article deals with a successful attempt to develop and test alternative designs of 380-kV power transformers, for the impulse testing of which, with 1/50 wave, the exceedingly high peak value of 1775 kV was prescribed. The severity of the demands on the electrical strength of such transformers can be visualized when it is considered that, for an air gap to withstand a potential of 1775 kV, the flash-over distance must be nearly 4 metres. Consequently, the length of the high-voltage bushing leading into the tank must be of the same order. However, between the entry to the winding and the earthed end, there is available only about 1 m, the majority of which is occupied by the metal of the winding and only a small proportion by insulation. If the entire space between the h.v. pole and earth were filled with insulating oil alone and if measures were not taken to control the electrical field or to subdivide the flash-over distance, it would hardly be possible to prevent 1775 kV from flashing over a distance of 1 m. Despite the insulating distance being reduced to less than half by the winding copper, it was nevertheless possible to cope with the test voltage by subdividing the total distance into a large number of smaller distances with effective field control and suitably formed insulating sections.

Fig. 1 shows a diagram of the design selected, which employs a simple concentric winding arrangement, in which the 14-kV winding *uv* is surrounded by the 380-kV winding, the entry to which is in the middle. The main insulation between the h.v. and l.v. parts consists of a sufficient number of layers of impregnated paper. The external cylinder of paper, to allow the h.v. lead to be brought out, is slit down one side and splayed open. The uniform reduction in the service voltage towards the top and bottom of the winding results in hardly any potential between the ends and earth in service and thus obviates the need for special insulation between the ends of the coil and the l.v. winding or the iron frame.

The disc-coil design of h.v. winding chosen in this case has the drawback that the voltage distribution over the winding at the start of an impulse is far less uniform than with a layer winding, and initially the greater part of the impulse voltage concentrates at the line end. As the charging wave progresses through the winding it causes brief high-potential differences to appear successively at points further inside. In contrast, in layer windings, the voltage distribution between the beginning and end is more uniform, on account of the powerful capacitive coupling of adjacent windings and layers. The control of the high stresses between the ends of the layers and earth, however, presents considerable difficulty. After carefully considering the pros and cons, it was decided to use the disc-coil arrangement. The adoption of special measures made it possible appreciably to improve its fundamentally unfavourable voltage distribution by increasing the capacitance between successive coils by appropriately dimensioning and finely grading the insulation between conductors and between coils, and also by extremely close coupling of the first coils to the h.v. entry.

Three different designs of winding for e.h.v. transformers were tried. They differ from one another in the method of potential control in the coils at the line end, also partially in the increased dielectric strength between successive coils. For the tests described below, the magnetic circuit, tank, l.v. winding, and the main portion of the h.v. winding were the same throughout.

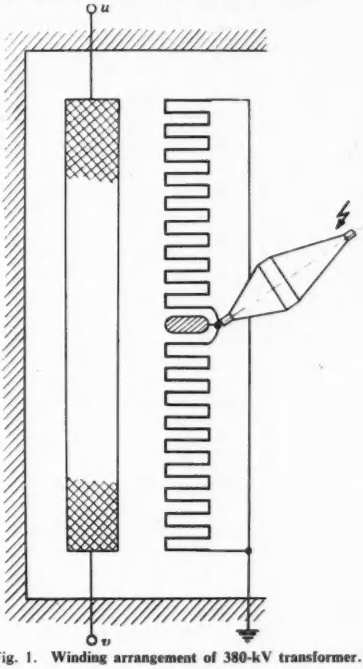


Fig. 1. **Winding arrangement of 380-kV transformer.**

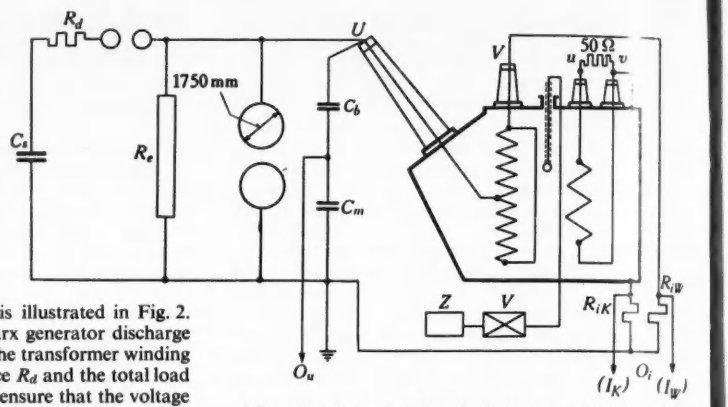
In each case, the length of single conductor from the entry to the end of the h.v. winding was about 5000 m. Assuming that the transient wave advances at a speed of about 150 m/μsec, the basic time for the wave to traverse the whole winding is over 30 μsec.

Impulse Circuit

Only the third and final winding was tested with a new 3·6-MV impulse generator rated at 100 kWsec, the two others being tested in the high-voltage laboratory, where the equipment has a full charging voltage of 2·4 MV with a nominal energy of 25 kWsec. Allowing for the load capacitance of the impulse circuit and the higher series resistance required, the maximum impulse voltage attainable with this older equipment is 1800 kV, so that it was just possible for the stipulated test voltage to be reliably generated. The generation of 1775 kV by the new equipment presents no problem ; in fact, a part of the equipment can be switched out in order to attain a higher impulse capacitance. In this manner it was easy to maintain the stipulated half-value time of 50 μsec.

Fig. 2. **Impulse circuit.**

U,V = Leads into h.v. winding.
u,v = Leads into l.v. winding.
C_s = Impulse capacitor.
R_e = Tail resistor.
R_d = Damping resistor.
C_b = Load capacitor.
C_m = Measuring capacitor.
O_u = Voltage-measuring system of the oscillograph.
O_i = Current-measuring system of the oscillograph.
R_{1K} = Resistor for measuring current I_K transferred to the core and l.v. winding.
R_{1W} = Resistor for measuring neutral current I_W at the end of the h.v. winding.
V = Amplifier.
Z = Oscillograph.



The impulse circuit diagram is illustrated in Fig. 2. The impulse capacitors in the Marx generator discharge through the tail resistance R_e or the transformer winding under test ; the damping resistance R_d and the total load capacitance of the impulse circuit ensure that the voltage rises, as desired, without oscillation, where possible in not more than 1·5 μsec. The load capacitor C_b determining the wave front, with a capacitance of the order of magnitude of 1000 pF, acts simultaneously as the means of reducing the impulse voltage to a level suitable for recording by the fast-sweep oscillograph. For this purpose a low-voltage capacitor with about 2000 times this capacitance is connected in series with it, the voltage experienced by this capacitor being fed over a screened cable to the oscillograph. The secondary winding of the transformer is bridged by means of a resistor, so that the voltage at its outlets is limited to harmless amounts.

To supervise the waveform of the surge and for the task of fault detection with high sensitivity, it is now more or less general practice to measure the impulse current, as well as the impulse voltage. The preferred method is to measure the current at the earthed end of the winding (" neutral current "). Sometimes, the current between the tank and earth is recorded. In this case, a shunt of low ohmic value is inserted in the earth lead, and the voltage across it is fed to another oscillograph circuit. With such important test objects as considered here, both currents are recorded, for in this way it is possible to obtain a definite indication of the nature of a possible fault.

In addition to this method, where the purely electrical quantities associated with the impulse are recorded, a highly sensitive method, particularly notable for its simplicity and, at the same time, for the excellent results obtained, has been developed by Brown Boveri. Known as the electro-acoustic detector, the guiding principle behind this equipment is that the pressure waves and vibrations set up in the transformer oil by a spark are picked up by a highly sensitive probe inside the tank, and the voltage fluctuations emitted are recorded after being suitably amplified. This no longer requires the high recording speed of the fast-sweep oscillograph, but permits an ordinary small oscillograph to be used.

These two methods of supervision, while quite distinct, supplement one another excellently. The oscillograms produced by the fast-sweep oscillograph, in addition to showing the entire course of the surge, also bring to light its every detail, as determined by the peculiarities of the test object or the remainder of the impulse circuit, and thereby enable conclusions to be drawn regarding the nature of a fault and its location. This knowledge is indispensable if appropriate steps are to be taken to overcome a disturbance. The detector, however, can only indicate the existence of a fault. It does this without any time-lag, because the spot of the oscillograph is observed while it is being photographed, and any deviation from the flat zero line clearly denotes the occurrence and the approximate extent of an insulation breakdown. Without first having to develop the film, the outcome of the impulse is known and the test can be continued to completion, reducing the time taken in testing. Naturally, the two-fold supervision increases the reliability of fault indication. If, for instance, the trigger of the oscillograph fails to function or its oscillogram is illegible, due to faulty manipulation, there is still the record of the simple and therefore very reliable detector, which is sufficient to assess satisfactorily the outcome of the test. It is not unusual for the method of recording the neutral current to be too sensitive, so that irregularities are indicated which have nothing to do with the winding insulation. For example, there may be low-power discharges at some quite harmless point inside the transformer, bearing no relation to the winding insulation ; such disturbances may perhaps be attributable to external corona phenomena, a poor contact somewhere along the measurement cable, or a faulty earth. As none of these affect the detector, harmless discrepancies of this nature can be distinguished from real insulation breakdown.

Winding Tests

A series of tests was conducted on three different windings. In the first test, the impulse front time of 1·45 μsec was just within the admissible upper limit ; the tailwave half-value time at roughly 50 μsec was almost exactly the standardized value. The oscillograms were found to follow the usual course for disc-coil windings : Apart from the slowness with which the initial oscillations died down, the neutral current remained practically at zero until the end of the basic time and only rose sharply when the charging wave arrived at the end of the winding ; the tank current during the first basic time was steady at the value $I = U/Z$ (Z being the characteristic impedance of the winding), the original, nearly square wave eventually becoming flattened and rounded while oscillating with a periodicity of double the basic time : after double this time, the impulse voltage diminished more rapidly on account of the increased load caused by the short-circuit at the end and now indicated at the beginning of the winding ; this sequence was frequently repeated.

Above 1100 kV, the current curves exhibited transitory disturbances, probably caused by pre-discharges. Without worsening the defect seriously, the test was continued to 1350 kV. The shape of the current curves remained completely unaffected—a sure sign that no inter-coil or

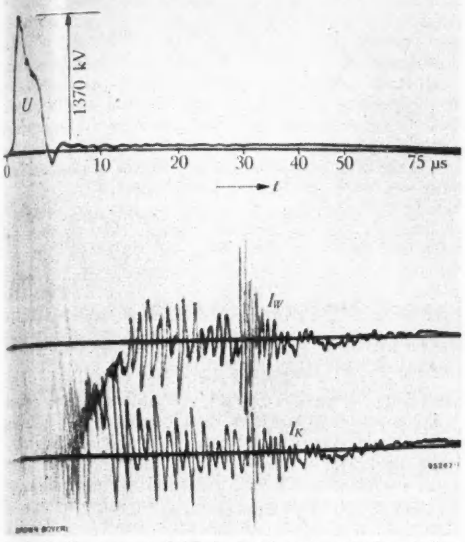


Fig. 3. Oscillograms recorded during impulse tests on the first winding.

inter-winding faults had so far occurred. With the subsequent shot, employing a slightly increased voltage and the opposite polarity, a serious winding fault occurred. The oscillograms showed that the voltage collapsed at the peak, with violent transient oscillation in the current ; the arc restruck at a time t between 29 and 30 µsec, probably near the end of the winding, as the narrow hump of the chopped voltage wave stress passed over it. Breakdown occurred at 1370 kV, as shown in Fig. 3, in which I_w is the neutral current, and I_k the impulse current transferred to the core and l.v. winding.

When the test rig was dismantled, there were no signs of discharge on the inner surface of the h.v. winding. However, on the outside, flash-over had taken place across one half of the winding, and the insulation at the beginning and end of this half was punctured, as were some of the intermediate sections. The cause of the defect was found to be the inadequate strength of the winding and control arrangement against the stresses experienced.

The transformer was reconstructed for the second test, using the main part of the h.v. winding, while the groups of coils on either side of the entry were replaced by specially formed coils with increased coupling.

Even at low voltages, the impulse currents, the tank current in particular, were distorted by superimposed high-frequency oscillations which died away rapidly. However, it may be assumed with reasonable certainty that these were due to sparking from an unearthed part of the iron circuit, and thus signified no danger to the transformer. It was unnecessary to remove the active part to try to overcome this minor disturbance. Furthermore, the indication of real faults was not drastically affected by these impulses.

The voltage was raised to 1620 kV without further complications, and not until the next shot at 1670 kV did an insulation breakdown occur in the heart of the winding. The basic curves of both currents subsequently changed after violent transient oscillations. The voltage curve also showed the fault by an immediate small dip and subsequent large discrepancies. These signs denoted that flash-over had taken place across a group of coils near the end of the h.v. winding. The entry to the winding was not touched and remained quite undamaged. Despite the existing damage, it was possible to attempt a further test of the winding, particularly the entry, with a chopped wave. In this case, chopping was effected by means of a rod gap of such a width that the voltage collapsed 3 µsec after the impulse began. Although the voltage was raised to the full capacity of the generator (about 1800 kV under the conditions obtaining), a large number of shots was performed without the slightest sign of a breakdown, as was indicated by a comparison of the oscillograms, especially of the impulse currents, and from the data supplied by the electro-acoustic detector. Occasional slight deviations in the current were experienced, but these were almost certainly caused by variation in the chopping time and from uncontrollable influences in the earthing system.

In the course of these chopped-wave tests, a total of 33 shots between 1600 and 1800 kV was applied to the transformer, eight of these being at the maximum value. A further increase in voltage was not possible, owing to the limitations of the 25-kWsec impulse generator.

When the transformer was dismantled, it was proved that the newly incorporated coils were quite undamaged and that the fault lay in the main part of the winding which had been taken over from the first test. It must be assumed that an insulation breakdown had not been properly repaired and was responsible for the defect described, which occurred with the full-wave impulses.

In order to exclude the possibility of a repetition of the above occurrence, a third test was carried out on a completely reconstructed h.v. winding, in which the entry part contained a newly developed control system designed to distribute the voltage stresses particularly effectively. In addition, the longitudinal strength of the winding was increased by supplementary means.

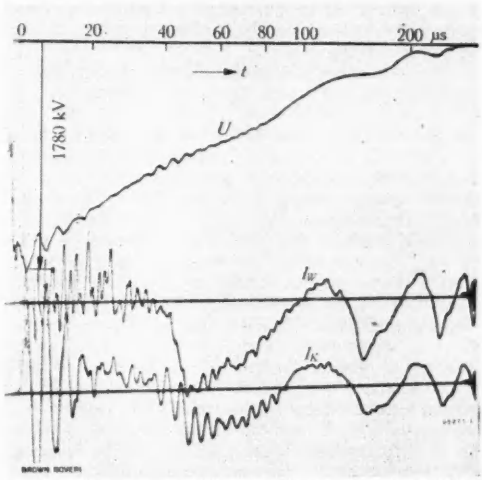


Fig. 4. Oscillograms recorded during impulse tests on the third winding.

After raising the impulse voltage in steps, the stipulated voltage of 1775 kV was reached without incident and the result confirmed by repeating several times. One of these oscillograms is reproduced in Fig. 4. As great care had been taken with the earthing of all the iron circuit, some parts of which had previously become charged in the electric field and produced low-energy sparks, the current curves this time were entirely free

from disturbance. The oscillations about the smooth mean value of the voltage curve, as can be seen in the initial stage of the oscillogram, are not due to inadequate damping of the circuit but have their origin in an internal transient process in the first part of the winding. They are also distinctly visible in the curve of the tank current.

Conclusions

Taking everything into consideration, the tests were, as expected, successful in that the extremely severe stresses imposed on the transformer were reliably mastered, and at the same time gave very valuable leads for possible future improvements. It should also be mentioned that these results have been confirmed by acceptance tests on the first transformer before delivery to the customer. In addition to withstanding three shots at 1775 kV and fulfilling all other conditions, this and the subsequent units were subjected to the induced voltage test at the extremely high figure of 780 kV (r.m.s.). These tests were also completed without incident.

JAPAN

The Corona-Resisting Properties of Polytetrafluorethylene

By K. Sugita, S. Nagao, and Y. Toriyama. (From *The Technology Reports, Tôhoku University*, Vol. 21, No. 1, 1956, pp. 179–183, 5 illustrations.)

Despite its superior mechanical properties and good thermal and chemical resistance, polytetrafluorethylene (P.T.F.E.) tape shows poor corona resistance. Tests were therefore carried out on P.T.F.E. sheets, 0·055 mm in thickness and 25 mm in width, to determine the mechanism of breakdown by corona. In the first instance, the sheets were mounted between a point electrode and a glass plate resting on a plane electrode, the point electrode, with a weight of 87 grams, merely resting on the P.T.F.E. At an effective voltage of 5 kV a.c., the onset of corona was observed, and the higher the voltage applied, the more intense was the corona. At 12 kV, a glow corona initiated around the point electrode, with a radially spreading corona terminating in fixed blue spots. After application of the voltage for 10 min, the P.T.F.E. sheets were examined microscopically, and numerous punctures, which had not been present before applying current, were found around the centre hole, these punctures corresponding to the fixed blue spots observed during application of the voltage.

From the above, it is considered that, before applying current, there might well have been a number of submicroscopic pinholes in the P.T.F.E. sheets, and that these holes were enlarged sufficiently by corona discharge to become visible during microscopic examination. Furthermore, microscopic examination frequently revealed a transparent liquid in the vicinity of the enlarged holes after breakdown. It is probable that this liquid is hydrogen fluoride formed by the chemical reaction between fluorine in the P.T.F.E. and moisture under corona discharge, but further investigations into this phenomenon were not pursued in the present study.

Subsequent tests involved the use of new, untested P.T.F.E. sheets and photographic plates, instead of the glass plates previously employed. After the application of an effective voltage of 6 kV for 1 min, many sensitized points, both inside and outside the central sensitized area, were noted. With cellulose acetate polymers, however, no sensitized points, other than the central sensitized area, were observed under the same conditions.

It should be noted that, when P.T.F.E. is heated quickly nearly up to the transition temperature (327°C) and then cooled rapidly, it becomes semi-transparent and exhibits a few fixed blue spots after the application of current. Microscopic examination in this case revealed only a few holes around the centre hole caused by corona discharge.

Additional tests were carried out on dielectric strength under various conditions, using sphere-to-plane electrodes in transformer oil, the radius of the sphere being 6 mm. These tests were conducted at different points, about 3 cm apart, on the same specimen, to avoid undesirable effects from preceding tests. It was found that heat-treated P.T.F.E. exhibited a higher dielectric strength than untreated P.T.F.E., probably because of the closure of most of the pinholes when P.T.F.E. approaches the transition temperature. To examine the effect of corona on the dielectric strength of P.T.F.E., the breakdown tests were repeated fifty times in air, using the same equipment and similar specimens, 0·055 mm thick. The results showed a mean breakdown effective voltage of 3·2 kV, a relatively low value when compared with the value of 5·6 kV obtained in oil, demonstrating that corona initiates more readily in air than in oil and results in a reduction in dielectric strength. Large variations in the dielectric strength of the specimens were found, with a mean deviation of about 1 kV, possibly because of the existence of pinholes and variations in thickness of the P.T.F.E. sheets.

Finally, P.T.F.E. was impregnated under vacuum with transformer oil and also with silicone oil, and in both cases the tests were repeated fifty times. Very little improvement was noted in the deviation, though the mean value of the dielectric strength was found to have increased. Accordingly, it may be assumed that there are tiny cavities in P.T.F.E., in addition to pinholes. Examination of untreated new specimens of P.T.F.E., utilizing an electron microscope (×6400), disclosed that the material had a granular structure and that there were numerous pinholes of the order of 0·1 micron. It would therefore appear that, in addition to pinholes and cavities, surface irregularities and the granular nature of the structure play an important part in the sensitivity to corona and variations in dielectric strength of P.T.F.E.

As a result of these investigations, it is concluded that the existence of pinholes in P.T.F.E. sheet is definitely established, and that sensitivity to corona and variations in dielectric strength are attributable to its granular structure and to these pinholes. It is also worth noting that, although heat treatment for a short time at about 327°C improves the corona resistance and dielectric strength of P.T.F.E., it is considered that some chemical reaction must also be taken into account as a contributory factor to the weakness observed under corona discharge, but further investigations in this connection are required. Moreover, the variations observed in dielectric strength present a number of problems which remain unclarified, including non-uniformity of the industrial product.

Dynamic Forces in Gear Drives

By J. ZEMAN. (From *VDI-Zeitschrift*, Vol. 99, No. 6, February 21, 1957, pp. 244–254, 24 illustrations.)

THERE have been considerable changes of view during the last twenty or thirty years on the loading of gear teeth and methods of calculating dynamic loads. It has been recognized for some time that the flank pressures derived from the Hertz equations are in most cases decisive as regards pitting and wear of gearwheels,[1] but the question as to why the permissible amount of flank pressure depends on speed of rotation or peripheral speed has been one on which various opinions have been expressed. On the Continent it was initially believed that this speed effect was associated with fatigue and that it was related to the Wöhler curve of the material[1,2,3], while in America it was considered that it was a result of transmission errors caused by vibratory forces.[4,5]

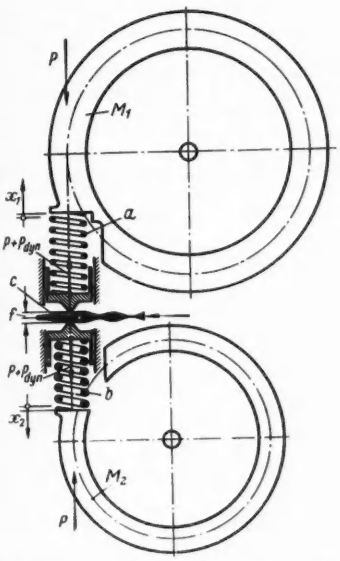


Fig. 1. **Equivalent system for a simple assessment of the dynamic behaviour of a pair of gearwheels.**

It has been shown by Niemann and Glaubitz[6] that, in the majority of cases, gearwheels operate within the limits of fatigue strength of the Wöhler curves, indicating that the "vibratory" or "dynamic" loads are the responsible factors. Investigations in this connection have been made by Buckingham[4,5] and by Niemann and Glaubitz.[6,7] The numerical results show considerable divergences, indicating that there is still some uncertainty regarding the basic laws governing the dynamic loads. Recently, a theoretical study of dynamic loads on gear teeth has been made by Tuplin[8], whose results, though interesting, give only something like a mean value, instead of the peak value of the load; moreover, the approximation method used does not give a general picture of the sequence of events. In the following investigation, an endeavour is made to represent the cycle of mechanical actions occurring under dynamic loading conditions.

TRANSMISSION ERRORS AND DYNAMIC LOADS

The measurement of tooth-profile errors and pitch errors is carried out by static gauging methods. For a pair of gearwheels in various stages of engagement, the total transmission error can be determined similarly and plotted against the angle of rotation or as a function of time for a given angular velocity. The second derivative of this curve, with respect to time, then gives the accelerations to which the wheels are subjected. These accelerations, which include the square of angular velocity as a factor before the sinusoidal terms, can be multiplied by the gearwheel masses to obtain dynamic forces. However, the forces thus derived are not the true dynamic forces, with which they would be identical only if the gearwheels were infinitely rigid. Gear teeth are capable of deformation under load; hence, they have a definite elasticity. The stiffness of the teeth, i.e., the ratio of force to deflection, is not a constant, but varies with load.[4,5,8,9,10,11] However, for simplicity, it is convenient to use a constant stiffness value in calculations.

At this stage, therefore, it may be concluded that (1) the errors determined by static gauging methods are not identical with the transmission errors occurring under running conditions, and (2) the dynamic loads cannot be determined only by these static errors, but require consideration of the stiffness of the gear teeth.

Since the gear-tooth stresses depend on vibratory deflections, it is necessary to determine the variation of the spring force of the gear teeth (i.e., the force resisting deflection) as a function of time, as well as the peak value of this spring force; in addition, it is necessary to establish the effect of running speed, angular velocity, or peripheral speed on these values of spring force.

EQUIVALENT MASSES AND SPRING OF GEARWHEELS

For a simple assessment of the dynamic behaviour of a pair of gearwheels, consider the system shown in Fig. 1. The two wheel masses are shown coupled together by means of helical springs (representing the gear-teeth stiffnesses) supported by plungers which are moved up or down by passing a strip or template of variable thickness between the contact points of the plungers. The strip represents the static engagement error of the two gearwheels; the springs are preloaded in accordance with the peripheral force transmitted. In this system, it will be sufficient to consider the mass and stiffness per centimetre of face width of each of the gearwheels (so that the units are kg sec^2/cm^2 for the masses M_1 and M_2, and kg/cm^2 for the spring stiffnesses C_1 and C_2). The error functions, i.e., the thickness variations of the template on either side of its zero-thickness line, will be denoted by f_1 and f_2, so that the total template thickness (and total error) is $f = f_1 + f_2$. Now, both the upper and the lower wheel masses and springs constitute vibrating systems, with an equation of motion $M_1\ddot{x}_1 = C_1(f_1 - x_1)$ for the upper mass and spring, and a similar equation (with subscripts 2 instead of 1) for the lower mass and spring. x_1 and x_2 are the displacements of the wheel masses away from their normal engagement positions when there is zero error.

As the forces on the two masses are equal ($M_1\ddot{x}_1 = M_2\ddot{x}_2$), it is possible to reduce the arrangement to an equivalent one-mass and one-spring system, which has displacements denoted by $x = x_1 + x_2$, under an impressed error function f. With the equivalent mass $M = 2M_1M_2/(M_1 + M_2)$ and the relation $M_1/M_2 = \ddot{x}_2/\ddot{x}_1$, it can easily be verified that the equation of motion for mass M_1 can be re-written as $\frac{1}{2}M\ddot{x} = C_1(f_1 - x_1)$,

and that for mass M_2 as $\frac{1}{2}M\ddot{x} = C_2(f_2 - x_2)$. These equations can be added together and, by substituting an "equivalent stiffness" $C = 2C_1C_2/(C_1 + C_2)$, the entire system can be represented by the single equation

$$M\ddot{x} = C(f - x) \quad .. \quad .. \quad .. \quad (1)$$

If damping is also taken into account, by means of a damping constant ρ, the equation becomes

$$M\ddot{x} = C(f - x) - \rho\dot{x} \quad .. \quad .. \quad (1')$$

The error function causing gearwheel vibrations will now be considered. Let z_1 and z_2 be the number of teeth of gearwheels (1) and (2) respectively, and let k be the highest common factor of these numbers ($k = 1, 2, 3$, etc.). Then, the same teeth of a pair of gears will come into contact again after $Z_2 = kz_2$ revolutions of gearwheel (1) and $Z_1 = kz_1$ revolutions of gearwheel (2).

Any errors related to this pair of teeth will therefore recur periodically, and for all the teeth the error function can be regarded as a periodic function which can be represented by a Fourier series, i.e., a sum of sine terms with frequencies which are multiples of ω_1/Z_2 or ω_2/Z_1, where ω_1 and ω_2 are the angular velocities of gearwheels (1) and (2). In practice, however, less general considerations appear sufficient, since the transmission error f occurs mainly in two forms, i.e., (1) as a transient error related to one pair of teeth, or (2) as a continuous error, occurring approximately to the same extent with all teeth.

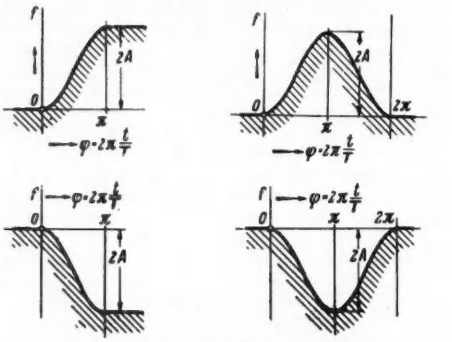


Fig. 2. **Examples of four main types of error curves, showing (above) positive errors and (below) negative errors.**

The transient error may be, for instance, a division error, which causes a variation of the pitch circle positions at a given instant ; after the disturbance, the wheels resume their uniform rotation. When the driven wheel is displaced forwards by the disturbance, the error is termed "positive" ; it is "negative" when the driven wheel is held back by the disturbance. Examples of four main types of error curves are shown in Fig. 2. Errors of these types can be represented by a portion of a sine curve, e.g., $f(t) = A_0 + A \sin[(2\pi t/T) + \gamma]$, where t = time, T = period, and γ = phase angle (in this case $\pm\pi/2$). The end of the disturbance occurs when $t = T/2$ or $t = T$ (half-wave or full-wave error). A steeply sloping disturbance curve corresponds to a small value of the period T of the disturbance, and a gradually sloping curve to a large value of T. Consideration of the transient error has been the basis of previous investigations into dynamic load effects.

Modern manufacturing methods endeavour to obtain uniform accuracy for all teeth in a gearwheel. The error then occurring is a continuous error, with approximately the same magnitude and the same variation for each tooth. In this case, the error function will be represented by a number of terms of a Fourier series, with a fundamental period $T = 2\pi/\omega_1 z_1 = 2\pi/\omega_2 z_2$. A harmonic analysis will give the amplitudes A_i and the phase angles γ_i of the sine terms forming this series.

Consequently, it will also be possible to represent the displacement x of the equivalent mass M by a Fourier series. The solution of eq. (1'), taking the first two terms only, can thus be written generally as

$$f(t) = A_0 + A \sin[(2\pi t/T) + \gamma] \quad .. \quad .. \quad (2)$$

both for the transient and the continuous error. However, the detailed solutions are different for the two cases. For the transient error, the solution should contain the maximum value of $(f - x)$, since $P_{dyn} = C(f - x)$ is the dynamic load on the gear teeth during the disturbance and a short time thereafter. In this case, the amplitudes are primarily determined by the disturbance, and the effect of the slight damping present can be neglected.

The continuous error, however, gives rise to a forced vibration, of which the steady-state (and not the transient) condition is of importance. In the resonance range, the amplitudes are limited only by damping, which cannot be disregarded in this instance. On the other hand, only the harmonic component corresponding to the natural frequency of the gearwheel needs to be considered.

The Transient Error

Let $C/M = \Omega^2$. Then, the general solution of eq. (1) is

$$x = -\Omega\cos\Omega t \times I_1 + \Omega\sin\Omega t \times I_2 + B\sin(\Omega t + \beta)$$

where $I_1 = \int f(t)\sin\Omega t \, dt$, and $I_2 = \int f(t)\cos\Omega t \, dt$, these integrals being evaluated between the limits $t = t_0$ and $t = t$, and β = phase angle of the free vibration. The term with the factor B represents any natural vibration already existing at the time t_0. This latter term can be neglected if the system is not vibrating at the time t_0, i.e., by writing $x = 0$ and $\dot{x} = 0$ at $t_0 = 0$. By substituting the expression for the transient error, i.e., $f(t) = A_0 + A \cos(2\pi/T)$ (with $\gamma = -\pi/2$) in the general solution, and taking account of the fact that $A_0 = A$ = half-total swing for transient errors of the cosine types considered,

$$x = \pm A\left[1 - \frac{1}{1-r^2}\cos 2\pi t/T + \frac{r^2}{1-r^2}\cos\Omega t\right] \quad .. \quad (3)$$

where $r = T_e/T$ is the reciprocal of the "tuning ratio" T/T_e, i.e., is equal to the period of the gearwheel vibration ($T_e = 2\pi/\Omega$) divided by the period T of the disturbance.

Eq. (3) gives the motion of the wheel masses during the period when the disturbance is acting. It may be noted that the movements caused by positive or negative transient errors differ only by the plus or minus sign of the result, i.e., in the direction of motion, as long as tooth contact is maintained by the peripheral force transmitted, as must be assumed in practice. Force reversals causing tooth separation give rise to excessive gear noise and must be avoided as far as possible ; hence, Buckingham's two formulae for impact after tooth separation, although of theoretical interest, do not come within the allowable range of practical requirements.

The expressions containing r^2 in eq. (3) are hyperbolic curves which reach infinitely high values for $r = 1$. For tuning ratios approaching unity, these expressions cannot be used in eq. (3), since they give an indeterminate result, i.e., $x = A(1 - \infty + \infty)$. By taking the limit of T tending towards T_e, however, x is obtained as

$$x = \pm A[1 - \cos 2\pi t/T_e - (\pi t/T_e)\sin 2\pi t/T_e] \quad .. \quad .. \quad (4)$$

The transient error occurs only over a period $t = T/2$ or $t = T$; when the disturbance ceases, the forced vibration of the gearwheel becomes a free vibration. For half-wave errors (of duration $t = T/2$), the free vibration is given by

$$x = 2A + B \sin(\Omega t + \beta) \quad .. \quad .. \quad .. \quad (5)$$

At the transition from forced to free vibration (i.e., at $t = T/2$), the value of the amplitude x must be the same with eqs. (4) and (5); the velocities $\dot{x}$ determined by the derivatives of eqs. (4) and (5) must also be equal at this instant. This enables the constants β and B to be determined, so that

$$\tan\beta = \frac{1 + \cos \pi/r}{\sin \pi/r} \quad \text{or} \quad \beta = \frac{\pi}{2}\left(k - \frac{1}{r}\right)$$

where $k = 1, 3$, etc., and $B = 2Ar^2 \cos(\pi/2r)/(1-r^2)$.

Similarly, for the full-wave error, the transition conditions at the time $t = T$ give

$$x = B\sin(\Omega t + \beta),$$
$$\beta = \pi[k-(1/r)], \text{ with } k = 2, 4, \text{ etc.},$$
$$\text{and } B = -2Ar^2 \sin(\pi/r)/(1-r^2).$$

A table of approximate B/A-values for different values of $T/T_e = 1/r$, is given below.

T/T_e =	0	0·5	1·0	1·5	2·0	2·5	3·0	3·5	4·5
Half-wave =	−2·0	−1·9	−1·6	−1·2	± 0·7	+ 0·3	0	−0·1	+ 0·1
Full-wave =	0	± 2·5	−3·2	± 1·6	0	± 0·5	0	± 0·2	± 0·1

In the range of values $T/T_e < 1$ (corresponding to a low natural frequency of the gearwheel, i.e., to a relatively low tooth stiffness, or to small values of T, i.e., to high wheel velocities or small pitch values), the absolute value of $1/(1-r^2)$ is smaller than that of $r^2/(1-r^2)$; hence, in eq. (3) the term with $\cos\Omega t$ is preponderant. As the frequency of the natural vibration is lower in this range than that of the disturbance, the wheel mass towards the end of the disturbance is still far away from the new equilibrium position, so that the free vibration then initiated reaches a high amplitude. Thus, high values of elastic deformation $(f-x)$ are obtained in the disturbance period and subsequently.

In the range $T/T_e > 1$ (relatively high tooth stiffness, or for large values of τ, corresponding to low wheel velocities or high pitch values), the coefficient $1/(1-r^2)$ is preponderant, so that, in eq. (3), the term with the disturbing frequency, $\cos 2\pi t/T$, is more important than the term with the natural frequency. In this case, towards the end of the disturbance, the new equilibrium position is already nearly reached; therefore, the amplitude of the subsequent free vibration is comparatively small.

Regarding dynamic loading, it should be borne in mind that extension of the spring (Fig. 1) represents, in fact, unloading of the tooth flanks, whereas compression of the spring increases flank pressure. The maximum compression, i.e., the positive maximum value of $(f-x)$, determines, therefore, in accordance with the equation $P_{dyn} = C(f-x)$, the additional dynamic load. This maximum value can occur, with transient errors, both in the disturbance period and later during the free vibration.

Whereas the motion of the wheel mass M differs for positive or negative transient errors only in regard to its direction, the magnitude of the dynamic load and also the instant at which it reaches its maximum value are different in the two cases. With a positive disturbance a compression of the spring occurs, whereas the spring is extended when the disturbance is negative.

By means of eq. (3), with $A_0 = A$, $\gamma = -\pi/2$, and the error function $f(t) = A_0 + A \sin[(2\pi t/T) + \gamma]$, the magnitude of $(f-x)$ is obtained as

$$f-x = Ar^2 \left[\cos\varphi - \cos(\varphi/r)\right]/(1-r^2) \quad .. \quad .. \quad (6)$$

where $\varphi = 2\pi t/T$. The maximum value is found by equating to zero the time derivative of this equation, giving the relation $\sin\varphi = (1/r)\sin(\varphi/r)$.

For convenience, the solutions of this equation can be represented in a graph of φ-curves plotted against the tuning ratio $1/r$, for the range of φ-values from 0 to 360 deg. It is found that the curves of the solutions are of the hyperbolic type, but with finite values at $1/r = 0$. The curves are evenly distributed throughout the entire graph. For instance, for $1/r = 6$, there are 6 curves intersecting the vertical up to 180 deg. (12 curves up to 360 deg.), i.e., there are 6 (or 12) possible solutions. Therefore, a restriction of the investigation to the spring deformation at the end of the disturbance[8] gives an incomplete picture. The maximum relative amplitudes $(f-x)/A$ obtained from eq. (6) for these various solutions are illustrated in Fig. 3.

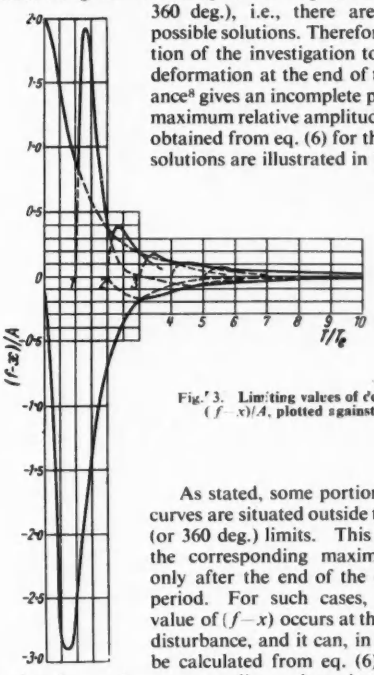


Fig. 3. Limiting values of deformation $(f-x)/A$, plotted against T/T_e.

As stated, some portions of the φ-curves are situated outside the 180 deg. (or 360 deg.) limits. This means that the corresponding maximum occurs only after the end of the disturbance period. For such cases, the largest value of $(f-x)$ occurs at the end of the disturbance, and it can, in these cases, be calculated from eq. (6) by substituting for φ the corresponding values, i.e., $\varphi = \pi$ or $\varphi = 2\pi$. However, it may be noted that the table of B/A values, previously given, provides all the desired information in such cases, since B is the maximum possible value of compression of the spring during this subsequent period.

In Fig. 3, the limiting values of the deformation $(f-x)/A$, which are decisive for the dynamic load $P_{dyn} = C(f-x)$, are given for the period of the disturbance, i.e., for φ-values less than 180 or 360 deg. (for half-wave and full-wave errors, respectively). The graph also gives $(f-x)/A$ values for the period after the disturbance, when φ can be taken as equal to 180 or 360 deg. Thus, the solutions in terms of φ-values, which can be numbered as 1st, 2nd, 3rd, etc., correspond to the curves with a positive peak at $1/r = 0$, a negative peak at $1/r = 0{\cdot}75$, a positive peak at $1/r = 1{\cdot}3$, etc. These curves overlap to a considerable extent.

In practice, all four types of transient errors may occur nearly simultaneously (positive and negative, full-wave and half-wave errors). It is then necessary to have

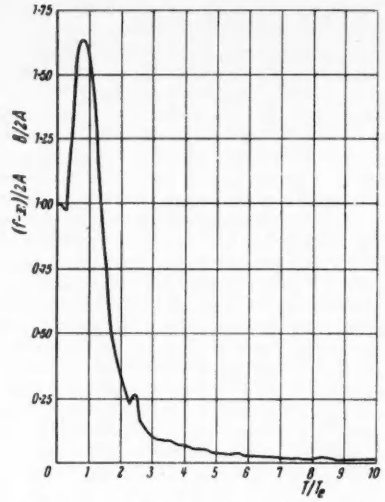


Fig. 4. **Maximum possible deformation $(f-x)/2A$ or $B/2A$ plotted against T/T_e.**

a single curve showing the maximum possible resultant values of $(f-x)$. Since A is the half-total swing, it is preferable to relate $f-x$ to the total swing $2A$. Such a curve for the maximum possible tooth deformation $(f-x)/2A$ or $B/2A$, for the disturbance period or the free-vibration period, is given in Fig. 4.

The Continuous Error

As the continuous error gives rise to a steady-state vibration, the term A_0 (occurring at the time $t_0 = 0$) can be made equal to zero by a suitable choice of coordinates. As a result, the error function can be written as $f = A \sin 2\pi t/T$. The particular solution of the differential equation (1') is then

$$x = a \sin\ [(2\pi t/T) - \alpha] \qquad .. \qquad .. \quad (7)$$

with $\tan\alpha = -(2\pi\rho/TC)/(r^2-1)$ and amplitude $a = A/[(r^2-1)^2 + (2\pi\rho/TC)^2]^{1/2}$. As usual, a non-dimensional damping factor $D = \rho/2\sqrt{MC}$ can be introduced to simplify these expressions. In the resonance range $r = 1$, the peak amplitude of vibration is then $a = A/2D$.

Regarding the values to be assigned to the damping coefficients ρ or D, and their dependence on the size, diameter, design, and arrangement of the gears, no investigation results are available. Some idea of the values can be obtained from the work of Glaubitz and Gösele[12], the resonance curves shown in their report corresponding to damping ratios D between 0·05 and 0·1. These values will be used as a basis in the following calculations.

The dynamic load, taking account of eq. (7), is

$$P_{dyn} = C(f-x) = CA\left\{\sin(2\pi t/T) - (a/A)\sin[(2\pi t/T) - \alpha]\right\} \quad .. \quad (8)$$

The maximum dynamic load is reached at a position φ, determined by $\tan\varphi = (a/A)\sin\alpha\ /[1-(a/A)\cos\alpha]$. The curve for $P_{dyn}/CA = (f-x)/A$ is a typical resonance curve, rising from unity at values of $1/r$ less than 0·1 to a peak of 5·1 for $D = 0{\cdot}1$ and $1/r = 1$, and falling off towards zero for increasing values of $1/r$ thereafter. Similar resonance curves are obtained for $D = 0{\cdot}075$ and $D = 0{\cdot}05$, with peaks of 6·6 and 10·0 respectively, at $r = 1$.

The curves for the relative amplitude a/A are typical "response" curves, rising from zero for $1/r$ values less than 0·1 to peak values at $1/r = 1$, and falling off towards unity thereafter. The peaks for the various D-values are approximately the same as (or very slightly lower than) those of the $(f-x)/A$ curves.

Application of Results to Specific Cases

The basic results of the investigations so far made are (1) the decisive influence of the tuning ratio T/T_e; (2) the existence of resonance ranges; (3) the possibility that the additional dynamic load may exceed the value of Cf; and (4) the dependence of dynamic load on the tuning ratio. There can hardly be any doubt about their validity, according to the theory on the manner in which dynamic loads are generated.

However, the immediate practical application and correlation with available data on the magnitude of dynamic loads are not yet possible, because these problems require knowledge of the true engagement error occurring in practice. Whether the usual gauging methods indicate this error correctly is an open question for various reasons; however, since at present there are no other types of data available, these measured values will have to be used to construct probable or at least possible types of error curves.

With these reservations, it may be noted that the transient or individual errors measured on gearwheels are as a rule steeper than the sine curve with a period corresponding to one pitch division. This, in eq. (2), can easily be taken into account by making T not equal to $2\pi/\omega z$, but relating it to a smaller value $2\pi/i\omega z = 60/inz$, where n is the gearwheel speed in rpm. The value of i cannot, as matters stand, be determined accurately. In the following it will be assumed that $i = 2$, but only for the purpose of showing how the results so far obtained can be applied to a numerical example.

The relations available for the tuning ratio are

$$T/T_e = 2\pi/i\omega z T_e = m\pi/viT_e = m\Omega/2vi = 60/inzT_e = 30\Omega/\pi inz \qquad .. \qquad .. \qquad .. \quad (9)$$

where m = module and v = peripheral speed.

The continuous error can occur, of course, in various forms, and generally the period of the fundamental vibration corresponds to the pitch. Not infrequently, the waveforms are of a sine-like shape; however, either the crest or the trough is longer than that of a true sine curve (Fig. 5). A harmonic analysis easily gives the first five terms of the Fourier series for the curve in Fig. 5 as

$$f = 0{\cdot}41\cos\psi + 0{\cdot}235\cos 2\psi + 0{\cdot}088\cos 3\psi + 0{\cdot}0195\cos 4\psi + 0{\cdot}00195\cos 5\psi$$

where $\psi = \varphi - \pi/2$, $\varphi = 2\pi t/T$ and $T = 2\pi/\omega t$, ω being the angular velocity.

The relationships between v, n, and $T/T_e = 1/r$ are given by the eq. (9). The additional dynamic force due to each harmonic under consideration can then be determined by means of eq. (8). In principle, it would also be necessary to evaluate the contributions of the flanks of all adjacent resonances to the resonance curve of the particular harmonic considered, but these contributions are small enough to be practically negligible. However, such contributions can build up to an appreciable value in the intermediate range of T/T_e values between two resonance peaks of two different harmonics. For a first

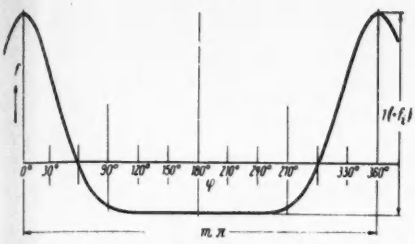


Fig. 5. **One possible type of continuous-error curve (idealized).**

approximation, the combined flank amplitude can be estimated by drawing in an envelope curve, as shown in Fig. 6. If this is not regarded as giving adequate accuracy, it will in all cases be sufficient to determine the contributions for one intermediate point in the range between two resonance peaks. As an example, a geared drive will now be considered with the following data, on which the curves in Fig. 6 are based :—Gear ratio 1 : 1 ; number of teeth $z_1 = z_2 = 30$; modulus = 1 cm ; wheel diameters $d_1 = d_2 = 30$ cm ; pitch-circle error (maximum permissible) $f_t = 0{\cdot}005$ cm. The tooth stiffness for steel and a flank angle of 20 deg. is, according to Buckingham[4], $C = C_1 = C_2 = 116{,}000$ kg/cm². (Buckingham's formula has been used because it is most convenient in this case.) The mass moment of inertia for a solid disc with a face width of 1 cm is $J_1 = J_2 = \pi d^4 w/32g$, where d = disc diameter, w = density, and $g = 981$ cm/sec². The mass relative to the pitch circle is therefore

$$M_1 = M_2 = M = J/(d^2/4) = 0{\cdot}00285 \text{ kg sec}^2/\text{cm}^2.$$

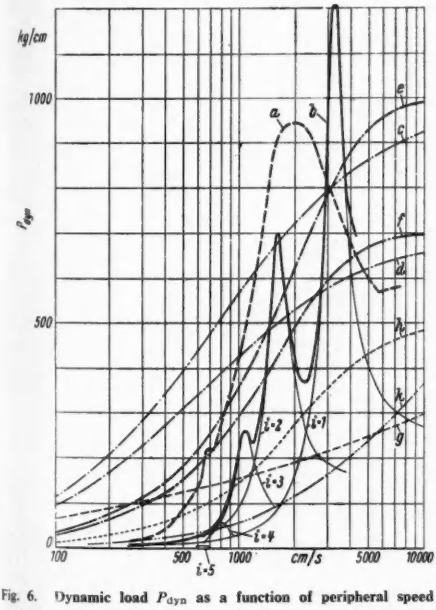


Fig. 6. **Dynamic load P_{dyn} as a function of peripheral speed V.**

(*a*) Curve for single disturbance of the type shown in Fig. 2.
(*b*) Curve for continuous error, according to Fig. 5 ; $D = 0{\cdot}1$; $i = 1$ to 5, according to Fourier series.
(*c*), (*d*) Curves based on Buckingham's old formulae ; peripheral forces = 420 and 120 kg.
(*e*), (*f*) Curves based on Buckingham's new formulae ; peripheral forces = 420 and 120 kg.
(*g*) Curve based on Niemann's new formula.
(*h*) Curve based on Niemann's old formula.
(*k*) Curve based on Tuplin's data.

Consequently, the natural phase velocity is $\Omega = \sqrt{C/M} = 6370$ rad/sec, and $T_e = 2\pi/6370 = 0{\cdot}000985$ sec.

For the transient error, let $i = 2$. Then, from eq. (9), $v = 3185/i(T/T_e) = 1593r$ cm/sec. Finally, with a modification of eq. (8), $P_{dyn} = Cf_i\,(f-x)/2A = 580\,(f-x)/2A$ kg/cm, this being the dynamic force per cm of face width. The values of $(f-x)/2A$ can be taken from Fig. 4.

For the continuous error, assuming a damping ratio $D = 0{\cdot}1$, the results for the various harmonics are as follows :— 1st harmonic : $P_{dyn} = 238(f-x)/A$; 2nd : $P_{dyn} = 136\,(f-x)/A$; 3rd : $P_{dyn} = 51(f-x)/A$; 4th : $P_{dyn} = 11{\cdot}3(f-x)/A$; and 5th : $P_{dyn} = 1{\cdot}13(f-x)/A$.

As previously stated, for $D = 0{\cdot}1$, the maximum value of $(f-x)/A$ is 5·1, and this value, inserted in the above expressions, gives the dynamic loads for the five harmonics, in kg/cm of face width.

For comparison, calculations for this pair of gearwheels have also been made using Buckingham's old and new formulae[4,5], the two formulae of Niemann[6,7], and Tuplin's data.[8] The various results are plotted in Fig. 6.

The extent to which the flexibility of the shaft carrying the gearwheel should also be taken into account also requires clarification. A good correlation of either the tooth/wheel-mass frequency or the shaft/wheel-mass frequency to measured values has not been found in attempts made by Glaubitz and Gösele[12] to determine the amount of agreement. The reasons for the discrepancies in frequencies have not been established.

It may be noted that the assumption of an invariable tooth stiffness holds with sufficient accuracy only for skew gears ; in this case the engagement is uniform in all possible positions of a tooth and the succeeding tooth takes the peripheral force to the same extent as it is removed from the previous tooth. This is not the case with spur gears or skew gears of small face width. At the beginning and end of the engagement of a tooth there are always two or more teeth in mesh, but at the middle of the engagement only one tooth is in mesh ; therefore, spring stiffness varies continuously. The equilibrium positions of the two different stiffness conditions occur with different periods, so that there are free decaying and superimposed vibrations. This subject has been investigated by Strauch,[13] but the method involves considerable computations.

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GERMANY

Straightening Processes for Cylindrical Parts

By M. MÜLLER. (From *Draht*, English Edition, No. 27, February 1957, pp. 42, 44, 46, 12 illustrations)

WHEN considering the problems involved in straightening cylindrical parts such as valves, push-rods, and shafts it is necessary to investigate the circumstances leading to the appearance of distortion during their manufacture. All these components represent forms in which the internal stresses are uniformly distributed about a neutral axis and are therefore in a state of equilibrium. Kinking may be due to many causes, the most common of which are heating or heat treatment and forming of the parts. Differences in heat stresses in the blank result in different internal stresses which will tend to compensate themselves by bending the neutral axis and consequently the workpiece, e.g., in the forming of a valve or the heads of push-rods. A similar effect is obtained if forming of the component results in an accumulation of material of varying volume, for example if the head is positioned eccentrically in relation to the shaft or if the parts are bent while being ejected from the drop-forging mould.

To eliminate the kink in the workpiece it is necessary to remove the internal stresses (e.g., by annealing); alternatively, new stresses must be set up artificially around the neutral axis in such a way that these stresses are of uniform magnitude at all points and parallel when the workpiece is straight. When using this latter method, it is clear that, irrespective of the cause of the initial deformation, the workpiece will have to be bent in order to straighten it. A bent valve, push-rod, screw, drill blank, etc., will be subjected to compressive stresses on one side, while its opposite side will be subjected to tensile stresses; it is the purpose of this straightening process to cancel the tensile stresses on the inside of the bend either by neutralization or by elimination. The part must therefore be bent in the direction opposite to that indicated by the direction of the bend by an amount determined by the elastic limit of the material. If the applied bending stresses are lower or within the elastic limits, the part will simply be subjected to a spring action and it will resume its initial position after the load has been removed from the part.

Fig. 1. Arrangement of straightening rollers in a disc-type straightening machine.

These remarks do not apply entirely if the load is applied in the form of a surge or impact, since, under these conditions, the elastic limits, as well as the yield point and flow limits, will greatly exceed their static values. The first straightening unit developed was as shown in Fig. 1, in which the workpiece was simply rotated in such a way as to enable all bends or arcs to come into contact with the straightening appliance. It was found, however, that rotation of the part led to the above-mentioned increase in the elastic limit, since the load was applied under impact conditions. Another disadvantage encountered was the fact that the point load due to the width of the rollers produced a notching effect on the workpiece. Apart from the purely external blemished appearance of the part due to this notching effect (usually removed during finish-machining), there was the danger that the material was being subjected to dead-rolling at these points, i.e., the notching at these points would result in a large local stress peak exceeding the ultimate breaking strength of the material, and finally leading to failure of the component. Practical experience has shown that this may quite easily occur with relatively brittle material and inexpert treatment. However, even in the absence of actual breaking failure, it was clear that, although the stresses required for straightening were of equal magnitude and evenly orientated, they were nevertheless very different in relation to the length of the component itself. Whether this fact has any bearing on subsequent machining or subsequent operating temperatures will have to be investigated. The disadvantages mentioned above could be overcome by barrel-grinding of the rolling tools. This would mean, however, that separate rolling discs would be required for every workpiece diameter, for every material, and for every pressure; the process would therefore be completely uneconomical.

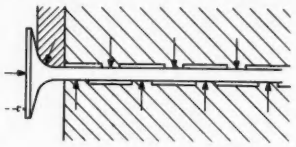


Fig. 2. Arrangement of straightening rollers and profile roller.

The object of further development lay in the elimination of the notching effect and its consequent disadvantages. This was achieved by means of a straightening roller consisting of a single unit provided with a trapezoidal thread. At first, great difficulties were experienced before the correct relationship between the tooth and gap of the trapezoid profile and the optimum pitch was determined. The profile of stationary straightening tools corresponds entirely to designs arrived at by theoretical considerations. As the rollers begin to rotate, the points of application of the forces begin to travel along the workpiece; dead-rolling of the material is therefore entirely prevented. Even small marks produced by the hobbing action of the threads are rolled away, and additional local stresses are eliminated. Tests have also shown that parts straightened by this process are less likely to suffer from distortion during subsequent operations than parts straightened by any other process. Distortion during subsequent treatment is usually caused by the removal of material from the surface (the greatest stresses are usually found here) and the consequent disturbance of the equilibrium about the neutral axis.

Straightening of valves presents the greatest difficulties, since the valve head must be straightened, in addition to the valve shaft. Straightening of a valve head is shown diagrammatically in Fig. 2. As before, the valve head must be bent beyond the elastic limit, and the valve rotated, to act on and remove distortion in every direction. The profile roller in Fig. 2 has a different diameter from that of the straightening rollers and covers a considerable range of diameters on the valve heads; this means that different surface speeds are generated, which, in turn, lead to considerable friction on the valve head. The line

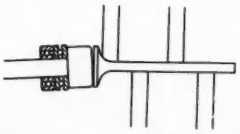


Fig. 3. Arrangement of straightening discs, profile roller, and thrust punch for straightening valves.

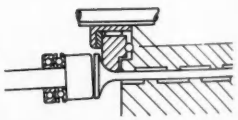


Fig. 4. Arrangement of straightening rollers, idling profile roller, and thrust punch for straightening valves.

of contact between the profile roller and the valve head had therefore to be kept as narrow as possible. This precaution has a lower limit, since an insufficient width of contact line leads to the appearance of undesirable surface marks (notching). To overcome this problem, the profile roller was arranged as a loose roller whose surface speed would therefore automatically adapt itself to the optimum value for the respective contact diameter.

It is clear that the two straightening processes (shaft and head straightening) have opposing effects if applied simultaneously, thus vitiating the result of the straightening process, a fact which has been proved in practice.

Since it is impossible to pre-calculate the rolling data (rolling pressure, rolling speed, and rolling time), in view of the unpredictable effects of the phenomena mentioned above, the effective range and application of the system have been demonstrated by means of operating data obtained in practice. In general, it is found that the rolling time should be reduced to the shortest possible period, i.e., approximately 1 to 3 sec, to avoid dead-rolling. With the tests made so far, the speeds selected varied between 30 and 40 rpm, while the straightening pressure for valves of 8·5 mm diameter and 120 mm length was in the region of 6 to 8 tons. The largest valves straightened so far had a comparatively small diameter of approximately

(*Concluded on page 173*)

U.S.A.

Improvement of Weld Joints by Roll Planishing

By H. L. MEREDITH and B. R. RUSSELL. (From *The Welding Journal*, Vol. 36, No. 2, January 1957, pp. 113–117, 9 illustrations.)

Loss in weld-joint strength can be attributed to four main factors, of which the foremost is loss in tensile strength, owing to the weld being a cast metal. The other factors are respectively loss in ductility, the notch effect caused by abrupt change in thickness at the weld edge, and the enlarged grain structure at the heat-affected zone ; the transition from cast structure to lamellar structure is not uniform.

To increase tensile and yield strength in the manufacture of sheet metal, it is necessary either to heat-treat or to cold-work the hot-rolled structure. In the case of wrought-type materials, which do not respond to heat treatment, cold working raises the ultimate tensile and yield strengths and improves ductility, if not overworked. Some metals which respond to heat treatment are also susceptible to cold working. For instance, the improvement in mechanical properties, particularly yield strength, is much greater in the case of 24S aluminium alloy (with a nominal composition of 4·5% Cu, 0·6% Mn, and 1·5% Mg), if the metal is cold-worked prior to artificial ageing. The small amount of stretching (about 1%) necessary to remove the buckles and distortion after quenching is sufficient to cause an increase in yield strength of double that developed by sheet which has not been cold-worked.

The objectives of roll planishing are manifold, the principal aim being to improve the mechanical properties of the weld joint. A typical example is that of the completed nose-section of a missile which failed during proof-testing on several occasions. In an effort to eliminate these failures, the longitudinal welds were hammer-planished to cold-work the weld, after which pressure tests showed no failures, the welds producing 100% joint efficiencies. Because, however, hammer planishing required skilled workers and was too slow for high production, a 10-ton roll-planishing machine was devised, in which two rollers, diametrically opposed, exert pressure at a small contact area, the top roller being the driver and the lower roller the follower. Extremely high or very low loads may be exerted on contact, the magnitude of the output load being adjustable from 0 to 20,000 lb. Hence, if a weld, $\frac{1}{4}$ in. in width, is to be roll-planished, a compressive force as high as 416,000 psi may be obtained if the driver makes a contact $\frac{1}{8}$ in. wide, using a force of $6\frac{1}{2}$ tons.

The correct application of the roll-planishing process to a given metal is dependent on six variables, i.e., (1) thickness of base metal, (2) weld width and contour, (3) height of weld reinforcement, (4) roll-planishing pressure, (5) roll-planishing surface speed, and (6) surface finish. Of these variables, the height of weld reinforcement and the weld width are the most important, and must be rigidly controlled during welding. For a given metal thickness, the weld width should be held to within a range of three to five times the base-metal thickness t, this ratio being in agreement with good joint-design practice and representing an economical balance. This rule of thumb applies to all metals, aluminium and magnesium being the hardest to control. Where filler metal is used, the height of weld reinforcement for both the weld-bead crest and root should be held approximately flush. Where no filler wire is used, and the weld joint is merely fused together, the weld-bead height above the base-metal surface is generally zero or concave. It is therefore necessary to control the concavity in metal thicknesses of 0·02 to 0·07 in., so that, where concavity is obtained, it should never be more than 0·01 in. deep. If the welding procedure is changed, producing convexity, the reinforcement should not exceed $0{\cdot}20t$ above the base-metal surface.

Other objectives of roll planishing are to relieve weld-shrinkage strain and also to flatten and smooth weld surfaces. Consequently, it is necessary to control the cross-sectional area of the weld. If the weld-bead crest height is greater than the base-metal thickness, it is easy to see that, when the excess weld metal is squashed back into the weld area, the adjacent base metal must move, causing excessive distortion and canning. The basic concept is very simple : Just enough weld reinforcement is required so that it may be compressed enough to refine the surface grain structure, counteract shrinkage strain, and be flattened.

The amount of cold working that a cast-weld structure may take is dependent on many factors. Roll planishing is limited only by the weld-joint configuration and metallurgical limitations of the base metal.

It is well known that long welds made in square butt joints will shrink transversely and longitudinally. The rate of longitudinal shrinkage is uniform when welding is carried out in a longitudinal fixture. However, the rate of

transverse shrinkage increases as the length of weld increases. Recent inert-arc butt-weld tests have shown that the average transverse shrinkage in 0·027-in. extra-hard Type 301 stainless steel was of the order of 0·010 to 0·012 in. in the as-welded condition. By roll planishing the welds at 40 psi on a standard machine, overall shrinkage was partially reduced. The average final shrinkage was reduced to 0·008 in.

To clarify this information, further tests were made on 0·025-in. full-hard Type 301 stainless steel, and consisted of welding six weld panels together. Each panel was sheared to a nominal width of $4\frac{1}{16}$ in. wide by 18 in. long and, after milling each edge square, the width of each specimen was measured, the average width being 4·0098 in. When each panel was laid side by side, the total width was 24·059 in. Five welds were made by inert-gas tungsten arc, fusing the square butts together ; no filler wire was added. Welding was effected in a production welding fixture incorporating the chill-shunt principle. The final dimension across the starting end of the welded specimen was 24·007 in., giving a total shrinkage of 0·052 in., or an average shrinkage of 0·0104, in. per weld. The final dimension across the finishing end of the specimen was 23·989 in., giving a total shrinkage of 0·070 in., or an average shrinkage of 0·014 in. per weld. Thus, in 18 in. of weld length, the difference in as-welded shrinkage between the starting end and the finishing end was 0·0036 in. per weld.

The welds were roll planished after taking shrinkage measurements. A gauge pressure of 40 psi was used, with a $\frac{3}{4}$-in. width flat area on the top roller face and a 6-in. radius contour on the bottom roller, this combination being necessary because the weld-bead crest was slightly concave. Each weld was given a single planishing pass. The final total measurement across the starting end was 24·028 in., indicating that planishing caused an average expansion of 0·004 in. per weld at the starting end. The final total measurement across the finishing end was 24·012 in., showing that (1) the average expansion was of the order of 0·0045 in. per weld, indicating uniform planishing ; and (2) a slight cumulative increase in expansion can be expected with straight-through planishing.

Tensile Strength of Planished Welds

Since roll planishing is relatively new in the field of light-metal fabrication, the question obviously arises as to what effects planishing has on weld strength. In this connection, it may be stated that roll planishing has been found (1) to increase the ultimate tensile strength, (2) to increase the yield strength, (3) to decrease slightly the elongation, and (4) to improve the fatigue strength.

Room-temperature tensile tests, made to compare as-welded tensile strength with roll-planished tensile strength, have been carried out on semi-automatic inert-gas tungsten-arc welds in Type 301 stainless steel and on 24S and 61S aluminium alloys, and the results have been most encouraging. The effects of processing procedure on the mechanical properties of inert-arc butt welds in 61S aluminium alloy (with a nominal composition of 0·25% Cu, 0·6% Si, 1·0% Mg, and 0·25% Cr), with and without roll planishing, are shown in Figs. 1 and 2, Fig. 1 relating to the " soft " 61S-O alloy, and Fig. 2 to the initially heat-treated 61S-T4 alloy. Starting from column (A) in each case, these graphs show that planishing helps to improve mechanical properties. Although not represented in these graphs, one of the most significant results was that planishing improves weld-joint efficiency. Weld-

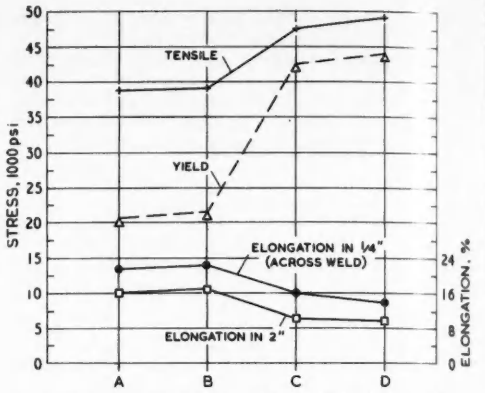


Fig. 1. Effect on mechanical properties of roll planishing semi-automatic inert-arc butt welds in 61S-O aluminium alloy.
(A) After welding, specimen heat-treated to T4 condition.
(B) After welding, specimen planished and then heat-treated to T4 condition.
(C) After welding, specimen heat-treated to T4 condition, and then aged to T6 condition.
(D) After welding, specimen planished, heat-treated to T4 condition, and then aged to T6 condition.

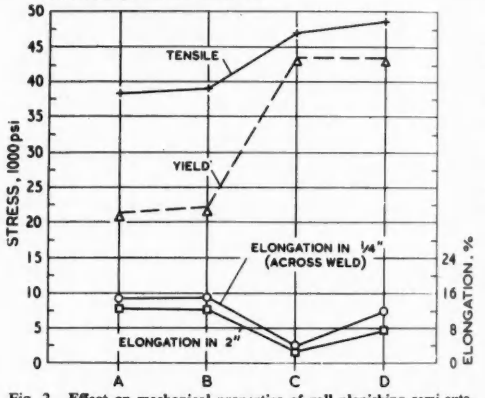


Fig. 2. Effect on mechanical properties of roll planishing semi-automatic inert-arc butt welds in 61S-T4 aluminium alloy.
(A) As-welded.
(B) Planished after welding.
(C) After welding, specimen aged to T6 condition.
(D) After welding, specimen planished, then aged to T6 condition.

joint failures were more predominant with the processing procedures in columns (A) and (C) than with those in columns (B) and (D).

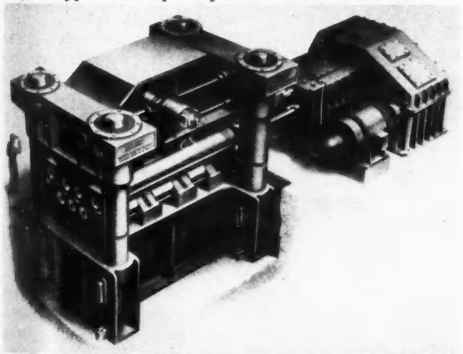
The most significant results of these tests are shown in Fig. 2, in which column (C) represents data obtained from welded specimens processed in the normally accepted manner. Here, it will be noted that the average elongation is approximately 2·7% for the base metal and 4% for the weld joint—a very low value, representing a dangerous condition, especially if notch-type stress raisers exist at the weld edge. In column (D), however, the data obtained were for specimens taken from the same welded panel as that used for column (C), the only difference being that the welds were planished before ageing to the T6 condition. In this case, it will be seen that, although the ultimate strength was only slightly improved (from 47,300 to 48,600 psi), and the yield strength remained fairly constant (43,600 to 43,500 psi), elongation was greatly improved (from 2·7 to 7·6% for the base metal, and 4·0 to 12·0% for the weld metal).

British Industrial Developments

Heavy Plate Leveller for Shipbuilding

A new roller leveller, capable of handling ships' plates up to a maximum of 2 in. thick and 12 ft wide to the standards demanded, particularly where welded construction is used, has been designed by The Head, Wrightson Machine Co. Ltd., of Middlesbrough. The first unit of this type, weighing approximately 200 tons, is now being built for Swan Hunter & Wigham Richardson. The machine is of very robust construction and measures 35 ft in length by 13 ft in width by 18 ft in height, of which 6 ft will be below floor level.

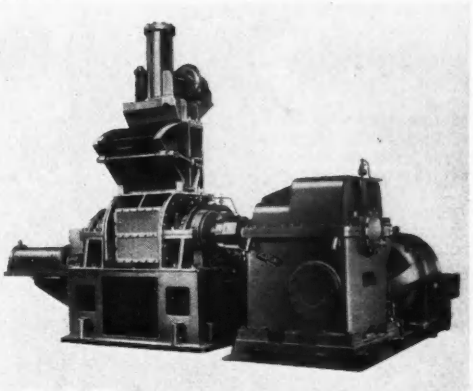
To enable the number of levelling passes required to be reduced to a minimum and to extend the range of thickness of plates which can be handled on one machine, the diameter of the levelling rolls in this new machine has been reduced as much as possible and, by using three banks of backing rolls across the face width of the working rolls, it has been found possible to use a roll diameter of 15 in. There are altogether seven levelling rolls, of which four are arranged in the top assembly and three in the bottom assembly. The two top outer rolls have a diameter of 18 in. and are supported, while the two centre top and three bottom rolls have a diameter of 15 in. and are supported by backing rolls of the same diameter, all roll adjustments being motorized. A main screw-down gear is provided for raising the whole of the beam carrying the four top rolls, and the two outer rolls have independently adjustable roll settings. The backing rolls are arranged so that, in addition to giving maximum support during the forward and reverse action of the leveller, they also allow scale to fall clear into trays provided in the base of the machine. All rolls are made from hardened alloy steel, and the various roll settings are shown on wide scale indicators. The top and bottom assemblies of the leveller are made up of unit steel castings forming the beams and side housings, and these are mounted on substantial forged-steel corner-posts incorporating worm gearing for adjustment of the top assembly, which is hydraulically balanced. The five centre levelling rolls are individually driven from a totally enclosed, triple-reduction gearbox through spindles of the universal-joint type. The top entry and exit rolls are idlers.



A floor-mounting contactor control panel houses the control gear for starting up, running, and protection of the equipment. Push-buttons and indicator lamps for the various controls are mounted on the front of the cubicle. The controls for occasional adjustments, such as change of speed and alteration of main-roll settings, are housed in a floor-mounting control desk. Duplicate control stations mounted on the entry and exit sides of the machine give the operator immediate access for fine adjustments of the outer rolls during levelling of the plate in either forward or reverse directions.

Spiral-Flow Intensive Mixers for Rubber and Plastics

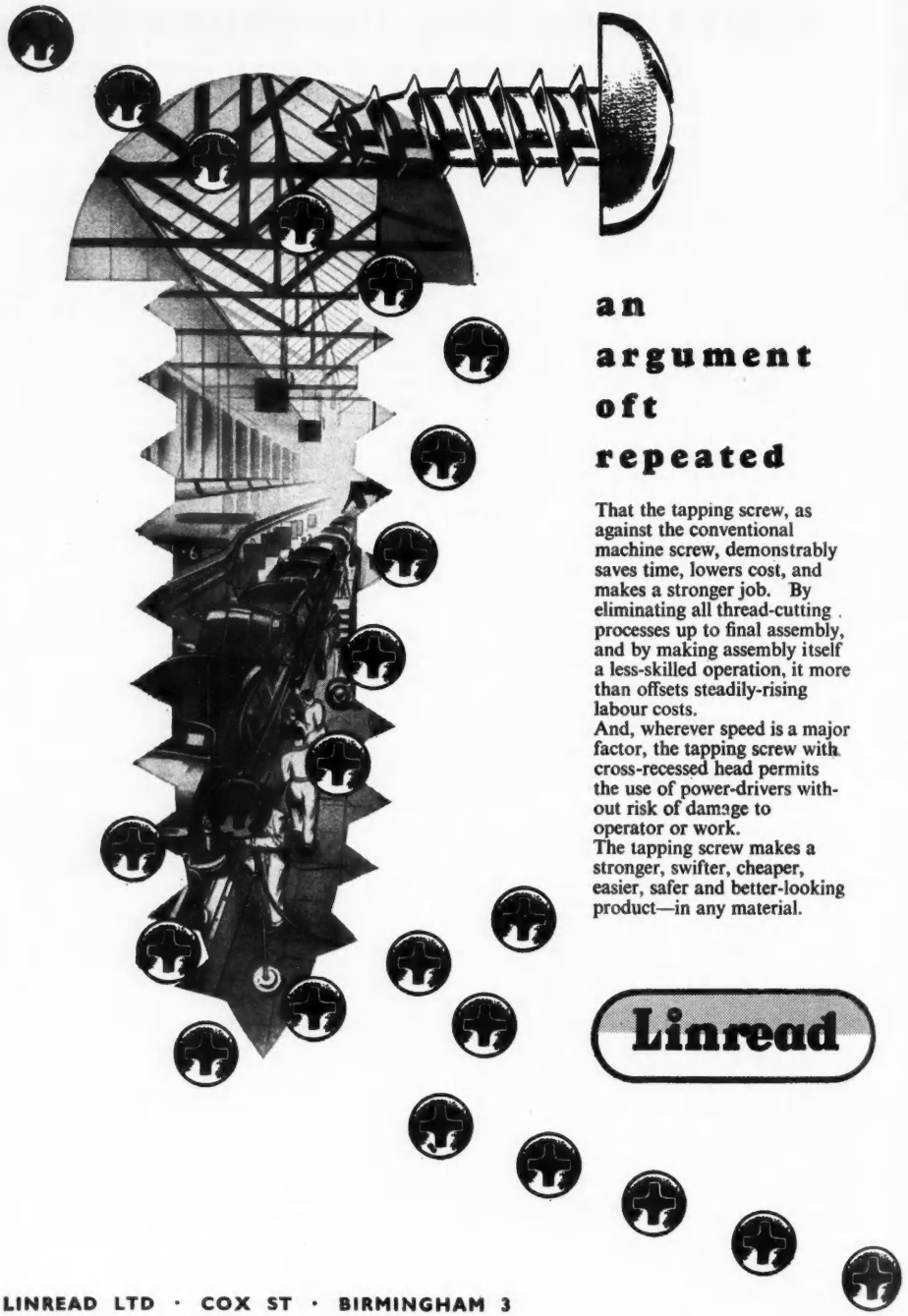
Following on the introduction of the Davis-Standard range of extruders for rubber and plastics, Fawcett-Finney Limited, of Birmingham and Bromborough, announce that a licence has been concluded with Stewart Bolling & Co., Inc., of Cleveland, Ohio, to manufacture and market the range of Bolling products in Great Britain, the Commonwealth, and Europe. This range of products comprises laboratory, experimental, and production units for rolls of from 6 × 13 to 22 in., and 26 to 100 in.; calenders of 2, 3, and 4 rolls capacity, for crepe, plastics, mastics, and gaskets; vulcanizers, from 20 to 36 in. standard, and 42 × 72 in. special, with lengths as required; and spiral-flow intensive mixers for rubber and plastics.



The Bolling spiral-flow intensive mixer (illustrated) excels on polythene and on butyl rubber, where temperatures much higher than normal are required; temperatures up to 400°F can be catered for, when flood lubrication is employed. Other advantages of this mixer include faster mixing and dispersion, longer effective life, lower maintenance costs, remarkable accessibility, and 20% less horsepower consumption, as a result of the specially designed rotor.

Fatigue-Testing Equipment for Aircraft Components

Equipment capable of completing within twelve months the entire component fatigue-testing programme for a major aircraft project is being installed by Short Brothers & Harland Limited. Already in use is a huge machine manufactured by Losenhausenwerk AG, of Düsseldorf, Germany. Hydraulically operated, it can impose a load of 100 tons as rapidly as ten times a second. Two smaller machines for testing materials are also installed, and the range will shortly be completed with the arrival of an Avery-Schenck machine capable of imposing loads of up to 20 tons on small jointed specimens. The Losenhausenwerk machine (illustrated) has already been used by Shorts to test aircraft components ranging from



an argument oft repeated

That the tapping screw, as against the conventional machine screw, demonstrably saves time, lowers cost, and makes a stronger job. By eliminating all thread-cutting processes up to final assembly, and by making assembly itself a less-skilled operation, it more than offsets steadily-rising labour costs.
And, wherever speed is a major factor, the tapping screw with cross-recessed head permits the use of power-drivers without risk of damage to operator or work.
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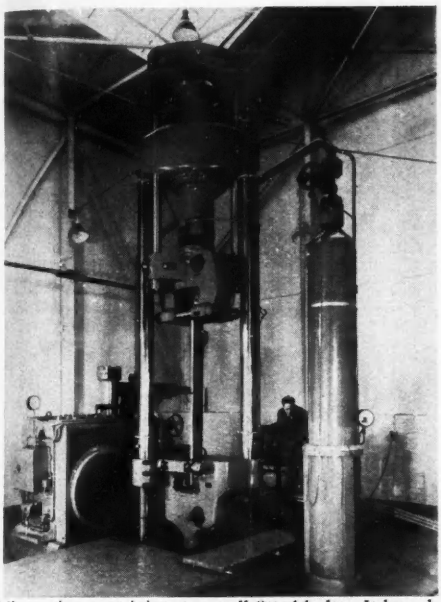


airliner wing-root joints to small fitted bolts. It has also been employed to investigate fatigue problems outside the field of aviation, such as the testing of specimens cut from a turbine disc used in a power-station.

The machine tests components by holding them between jaws which apply load pulsations of varying capacities. To accommodate parts of different sizes, the lower head is adjustable by means of an electrically actuated screw. In this way, components up to about 8 ft long can be tested. Loads can be maintained throughout protracted tests within a tolerance of 0·25 ton, which represents the change of load necessary to open and close a pair of electrical contacts. When the machine has warmed up, the load drift is negligible, but it may increase slightly if excessive variations of temperature cause changes in oil viscosity and volume. Decreasing load, however, is corrected by the controls and, if it should increase beyond a predetermined value, the machine will automatically stop. If, through failure of the specimen or any other cause, the load should fall at such a rate that the pumps will not counteract it, the automatic control will again stop the machine. To ensure that no vibration is transmitted to other installations nearby, the machine is mounted on an independent concrete bed resting on springs.

General-Purpose Electronic Digital Computer

Designed for use in solving a wide range of mathematical, engineering, and scientific problems, a general-purpose, high-speed digital computer has been developed by Metropolitan-Vickers Electrical Co. Ltd., of Trafford Park, Manchester. The new computer, known as the "Metrovick 950", utilizes the binary notation and consists of five main parts, i.e., input equipment, storage, arithmetical circuits, output equipment, and control panel.

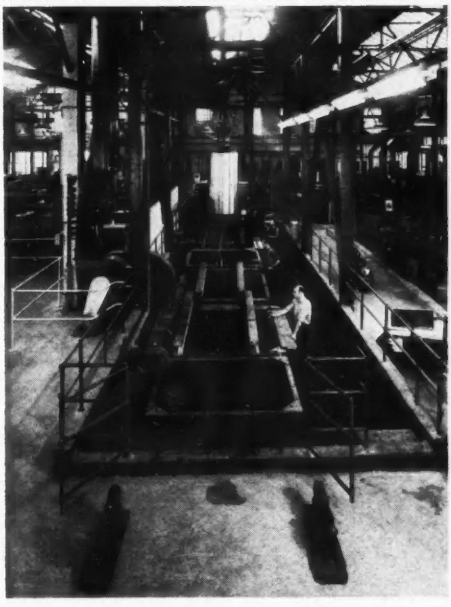
Information is fed into the machine by means of a photo-electric punched paper-tape reader, the paper tape having a five-hole code; the instructions and data are stored in a drum coated with a magnetic oxide, on which information is recorded in the form of magnetized spots on circumferential tracks. The arithmetical circuits per-

forming the calculations consist mainly of transistors. The output unit punches holes in paper tape under the control of the computer, and this information can subsequently be read or printed out on a teleprinter. The operator coordinates the whole machine, using the control panel, in such a way that instructions are obeyed correctly and in the right order. The Metrovick 950 is of simple and compact design, the overall dimensions being 14 ft 6½ in. long by 4 ft 10 in. deep and 6 ft 1½ in. high.

New Heavy Chromium-Deposition Plant

New plant for the electro-chemical deposition of chromium, capable of accommodating components up to 15 ft in length and 5 ft in diameter, and weighing up to 6 tons, has just been completed at the Port Glasgow works of Fescol Ltd., of London, N.7.

The runway over the five deposition and preparatory tanks has been designed to carry a load of 10 tons, so that no structural alterations will be necessary if the weight-lifting capacity should be increased at a later date. The vats are housed in an asphalt-lined pit excavated from



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shale and rock, with the tops on a level with the factory floor. Space has been left between each tank for inspection and repair, access for the operators being by means of a gallery along the length of the pit. The depth of excavation necessary brings the effluent from the vats below the level of the existing drains, so that a sump had to be formed in the gully, and acid-resistant pumps installed to transfer the effluent to the main drainage system. A specially designed system of cooling is used to ensure close control of the temperature of the deposition vats, and enables a heat-balance of $\pm 1°F$ to be maintained.

Current to the new plant is supplied via Westinghouse rectifiers and reverse switches. Two groups of these oil-immersed rectifiers, with stepless regulators, are placed close to the vats to ensure a minimum voltage drop in the conductors.

Heavy-Duty Face-Plate Coil Winders

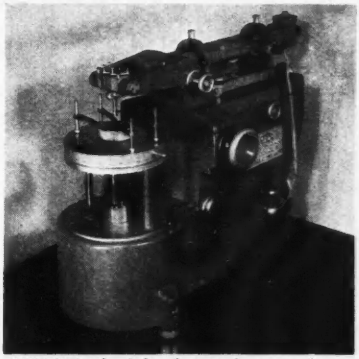
Designed for the winding of coils used in lifting-magnets and transformers, the latest addition to the extensive range of coil winders manufactured by Westool Limited, of Bishop Auckland, Co. Durham, is a heavy-duty face-plate winder. The machine, which is provided with a 30-in. face-plate, slotted to enable coil formers to be clamped to it, is capable of winding coils up to 28 in. in diameter with a maximum width of 6 in. The wires which can be used range from 10 to 30 swg.

The winder is fitted with a gearbox giving four speed ranges up to 415 rpm, gear-changing being effected by rotation of a spider handwheel. An infinitely variable ratio, within the limits of the machine, of the turns per layer is provided, with a calibrated scale for setting. The reel carrier and tensioning device are mounted as an integral part of the machine. Reels up to 15 in. in diameter can be carried, and adjustable tensioning facilities are included. A 2-hp motor and electromagnetic clutches and brakes are incorporated in the drive, ensuring smooth acceleration and deceleration and variable braking, from a smooth cushioning effect to an instantaneous stop.

Internal Measuring Machine

Added to the Matrix range in order to satisfy the demand for an internal measuring machine capable of measuring the diameter of very finely ground or lapped bores to an accuracy of $\pm 0{\cdot}00001$ in. or even finer, the new NPL/Matrix 3-in. machine announced by the Coventry Gauge & Tool Co. Ltd., of Coventry, is basically a comparator, the reference standards used being built up in the form of a box from high-precision slip-gauges and lapped end-pieces.

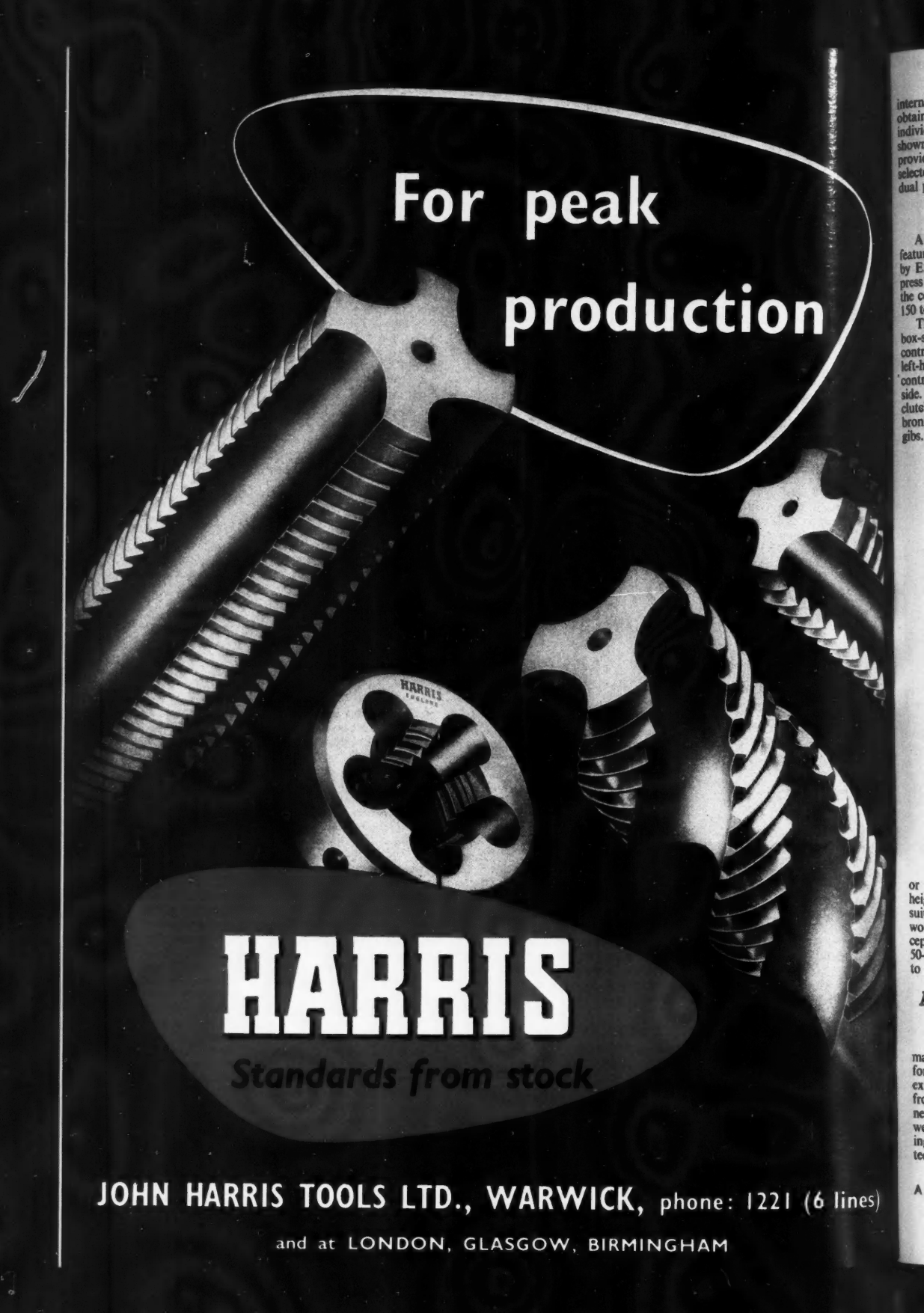
The new machine is capable of measuring internal diameters ranging from 0·1 to 3 in. to an accuracy of $\pm 0{\cdot}00001$ in. Diametral measurements can be made to this accuracy down to a bore depth of 1 in. and, by reversing the bore, end for end, on the work-table of the machine, this depth can be doubled. Actually, the machine will measure diameters down to a bore depth of 3 in. (or more on reversal), but over this extended range measurements must be made at lower magnifications, with a corresponding proportional drop in accuracy.



The NPL/Matrix 3-in. internal measuring machine consists essentially of a main casting supporting an elevating work-table and an adjustable slide which, in turn, supports as a single unit a Watts autocollimator and a measuring head. Longitudinal motion parallel to the axis of measurement and transverse motion can be imparted to the slide by means of handwheels and, after the slide has been set, it can be locked to the main casting by a hand clamp. The reference standard and the work to be measured are alternately placed on the work-table, which can be raised or lowered through a range of 6 in. by means of a handwheel, and finally locked by a clamp. The upper surface of the work-table can be rotated by hand through 360 deg. and, in addition, a fine rotational adjustment over a limited range can be applied to it at any position. Concentric circles are scribed on the surface of the work-table as an approximate guide for placing the standard and the work symmetrically in position, and quick-action friction clamps are provided for finally securing them in place. The edge of the work-table is graduated at 5-deg. intervals as a guide to the orientation of the measurements made. A stem, graduated at 0·05-in. intervals and rigidly attached to the work-table, indicates the axial depths of the diametral measurements made in the bore.

The measuring-head unit comprises a pair of optical reflecting mirrors mounted on a vertical arm, to which is attached a double-ended stylus which enters and contacts the bore under measurement. The double-ended stylus can be biased to contact each end in turn of the diameter being measured by applying small clockwise or counter-clockwise rotations respectively to a handwheel. Each mirror is capable of being tilted in the vertical and horizontal planes by rotating knurled screws situated immediately above the mirrors. The centres of these mirrors lie approximately on the axis of rotation of the measuring head. When correctly adjusted, each mirror respectively reflects the image of the autocollimator crosswires into the eyepiece, when the appropriate end of the stylus is contacting the bore under measurement.

The reference standard for the machine is of box form and is built up so that the internal dimension of the box is within about $\pm 0{\cdot}0005$ in. of the size of the bore under measurement. It comprises two lapped steel end-pieces which are separated the appropriate amount by two combinations of slip-gauges of equal length, the whole unit being lightly held by an exterior fitting clamp. The



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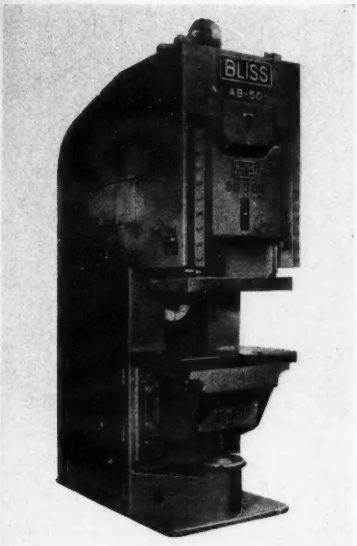
internal size of the composite reference standard is obtained by the addition of the known sizes of the individual separating slip-gauges, and experience has shown that this size can be relied on to within 0·00001 in., provided that slip-gauge combinations of equal length are selected and that care is exercised in bringing the individual parts together.

Adjustable-Bed Presses

A newly designed line of adjustable-bed presses, featuring a fully enclosed box-type frame, is announced by E. W. Bliss (England) Ltd., of Derby. The 50-ton press illustrated is the first size to be manufactured, but the complete line includes capacities ranging from 50 to 150 tons.

These presses are fully enclosed, with a box-crown, box-slide, and a completely built-in control system. Air controls and the recirculating oil unit are enclosed on the left-hand side of the press, and the press and motor controls are similarly flush-mounted on the right-hand side. Other features include a fast-acting air friction clutch and brake, an adjustable rotary limit-switch, and bronze-bushed bearings at the frame, connections, and gibs.



The 18 × 24 in. bed of the 50-ton model can be raised or lowered a distance of 9 in., making possible a shut-height adjustment of from 11 to 20 in., so that it is suitable for a variety of unwieldy stamping and forming work. The design of the welded steel frame, while exceptionally strong and rigid, requires little floor space, the 50-ton model being only 37 in. wide and 74 in. from front to back.

Automatic Equipment for Welding Oil-Storage Tanks on Site

Automatic girth-welding equipment is now being manufactured by Quasi-Arc Limited, of Bilston, Staffs., for the simultaneous welding *in situ* of internal and external circumferential seams of oil-storage tanks of from 30 to 200 ft in diameter. The time taken with this new equipment is much less than with normal manual welding methods. Automatic circumferential seam welding on such large vessels has hitherto rarely been attempted, because it is not usually practicable or economical to construct suitable manipulative equipment such as roller beds (normally used for rotating cylindrical vessels when welding) for such heavy and large-diameter work. Specially made equipment for use in construction on site would be difficult to transport, owing to its excessive weight and size. The same difficulties would also apply to transporting large prefabricated sections to the site.

With the girth-welding equipment, however, no manipulation of the vessel is required for circumferential welding. The tank is constructed vertically, strake on top of strake, instead of horizontally, as is usual with smaller vessels. The equipment consists of two Unionmelt DSH welding heads, with control panels and platforms for two operators. These are mounted on a saddle-type carriage which straddles the top strake of the tank. The drive and idler wheels of the carriage run directly on the curved edge of the plate, and guide-rolls are provided to act as a pivoting device. Thus, there is one welding head inside the vessel and one outside, with an operator on the platform of each. As the carriage moves at the selected welding speed round the perimeter of the tank, the internal and external seams are welded simultaneously in the horizontal position.

Vertical adjustment for the welding heads allows for plate widths from 4 to 9 ft, with additional local adjustment of 12 in. for accurate positioning of the welding heads. A spot-light guiding device is also fitted, to enable the operator to guide the welding nozzle accurately along the joint. The carriage movement is remotely operated from the control panels, and speed indicators are fitted. Holders are provided for oxygen and acetylene cylinders, and torches for pre-heating the joint ahead of the welding nozzle. Telephone sets for constant communication between the two operators are also provided.

The Unionmelt welding composition is supported under the welding nozzles by an endless-belt arrangement, tilted so that a weight of composition presses against the joint. Support is thus provided for the molten weld metal, so that welding currents up to 750 A can be used. A continuous melt-circulation system recovers the unfused composition and feeds it back to the hopper at the top of the machine. It then falls by gravity back to the welding nozzles. As each strake is completed, removal and repositioning of the equipment for the succeeding strake is simple and rapid.

The equipment can be used for plate thicknesses from $\frac{1}{4}$ to 3 in., or even thicker plate in some cases. The variable speed of welding ranges from 20 to 45 in. per minute, depending on the thickness of plate and welding current employed. Using $\frac{1}{8}$-in. diameter wire at 500 A, the metal deposition rate for each welding head is approximately 0·25 lb per minute, i.e., five times the rate of comparable manual welding. The deep penetration of Unionmelt welding also minimizes joint preparation and reduces the number of passes to about one-quarter of the number required in manual welding, resulting in faster welding with minimum distortion. The principle may, of course, be extended to many other types of structure.

Production of Synchronizing Rings by Powder Metallurgy

A new and economical method of producing high-strength synchronizing rings for automobile transmission systems by powdered metallurgy has been introduced by Machine Products Limited, of Swindon, Wiltshire. The new process enables complex contour components to be produced to such a high degree of accuracy that many machining operations hitherto essential are no longer required. Of the six such operations previously necessary to produce synchronizing rings, four have now been eliminated, and the only machining still required to complete the part is slotting and cutting a thread on the inside of the ring.

Made from a 10% tin bronze, these synchronizing rings have an ultimate tensile strength of 18 to 22 tsi, a

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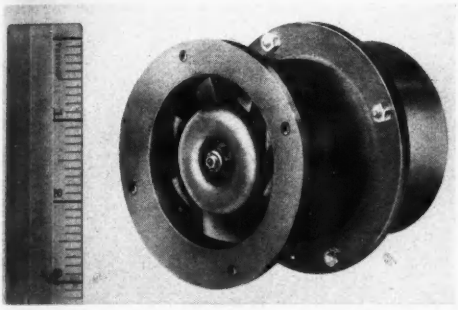
minimum elongation of 35%, and a hardness of 70 to 90 V.P.N. with a 5-kg load. Extensive laboratory tests and road tests in a number of different vehicles over the last two years have established that rings made by the new process are in every way as satisfactory as parts manufactured by conventional machining methods. Following on the successful completion of these tests, an initial production run has just been commenced.

Miniature Axial-Flow Fan

A very small 400-cps axial-flow fan of high performance has been produced by Woods of Colchester Ltd., for airborne applications, chiefly for cooling electronic apparatus. The fan is of the downstream guide-vane type, with an impeller of 2·4 in. diameter and an overall length of 3⅝ in., the weight being under 1 lb. The fan motor, a totally enclosed squirrel-cage induction 4-pole design, has a diameter of 1·6 in. Running at 11,000 rpm, it consumes 15 W and has an output of 30 cfm at a static pressure of 0·8 in. w.g. A 22,000-rpm version at present in course of development will have the remarkably high output of 50 cfm at 3 in. w.g. for the same weight.

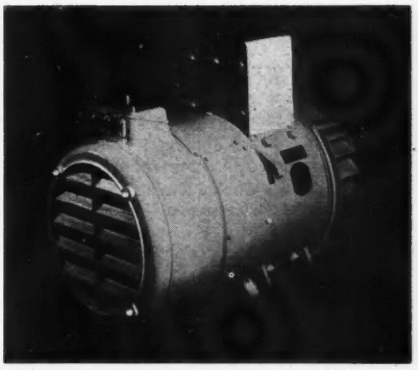


The guide vanes and motor carcase are integral with the casing, the whole being a one-piece aluminium pressure die-casting. The impeller is a plastic moulding with aerofoil-section wings. The rotor shaft is carried in single-shield ball bearings at each end, the bearings being spring-loaded axially to ensure quiet running. The motor is rated for 1000 hr continuous operation in an ambient temperature range of −40°C to +80°C. Owing to the efficient cooling provided by the integral guide vanes, the temperature rise of the windings and ball bearings is unusually low.

Alternator with Extremely Close Regulation

Of completely new design and differing from the standard Stamford alternator by a special control-box built on the top of the alternator, the Newphase alternator announced by Arthur Lyon & Co. (Engineers) Ltd., of London, W.1, is a machine which provides the closest regulation for specialized uses, such as telecommunication, electronics, X-ray work, etc. This additional voltage control serves to convert from a compensated machine to a closed-loop system with a positive voltage reference, against which the output voltage can be compared.

The control is made up of three main sections, i.e., (1) an error-detecting arrangement; (2) a magnetic amplifier for magnifying the power of the error signal; and (3) the control arrangement connected into the main excitation circuit. The first section utilizes a cold-cathode discharge tube as the non-linear element of a four-arm bridge, which is fed from the rectified output voltage of the alternator via an auto-transformer, stepping it up to a suitable value. The output of this



bridge is then fed to the control winding of the magnetic amplifier, which forms the second stage of the circuit, the power gained by this amplifier being in the region of 5000:1. Finally, the output of the magnetic amplifier is rectified, and the voltage so obtained is used to feed the control winding of the saturable reactor connected to the primary circuit of the voltage transformer.

Whilst the excitation adjustment for lead change is still brought about by the combination of the voltage and current transformer incorporated in all Stamford machines, any departure from nominal voltage occasioned by other causes is automatically corrected to within close limits by this overriding control.

Straightening Processes for Cylindrical Parts

(Concluded from page 167)

20 mm, a head diameter of 100 mm, and a length of 380 mm. The rolling pressure required was 25 tons. Straightening could be effected only by greatly increased rolling times. These data refer to tests carried out with disc-type straightening rollers (Fig. 3). The second method, using straightening rollers with a trapezoidal thread (Fig. 4), was used for straightening valves having shafts of 8 mm diameter, heads of approximately 30 mm diameter, and an overall length of 140 mm. Straightening was carried out at a rolling pressure of 6 tons, at a speed of 30 rpm, and a rolling period of 2 sec, i.e., under rather similar conditions to those with the first method. The straightening accuracy obtained with both methods was of the order of 0·1 mm for both the shaft and the head. This assumes that maximum distortion before rolling does not exceed approximately 0·8 mm, but these limits are in any case only rarely exceeded in actual practice. These data do not apply to very large valves of 380 mm overall length and 100 mm head diameter. With this machine, the output of the smaller valves mentioned above, straightened by this second system, was of the order of 400 parts per hour, using hand-charging. With a fully automatic valve-straightening attachment, outputs of 1200 parts per hour were obtained.

The accuracy obtained with the machine, when straightening push-rods of up to 380 mm length, was 0·15 mm ; the amount of distortion before straightening varied between 0·5 and 1·5 mm. In this connection, it should be mentioned that the heads of push-rods are sometimes displaced eccentrically in relation to the shaft, so that, even with a straight shaft, the axis formed during subsequent operations does not correspond to the neutral axis, resulting in a locked-in bending stress. Such eccentricity can hardly be corrected by rolling.

THE WELDER

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ADDITIONAL ABSTRACTS

Numerous important abstracts which, for lack of space, cannot be published in our main editorial columns are regularly included in condensed form in this section. Subscribers may obtain photostat copies of all original articles at cost.

Efficiency of Supersonic Nozzles for Rockets and Some Unusual Designs.

By R. P. FRASER, P. N. ROWE, and M. O. COULTER. (From an advance copy of a paper submitted to *The Institution of Mechanical Engineers, London*, March 1957, 19 pages, 30 illustrations.)

ALL rocket missiles and many rocket motors used to assist aircraft take-off can be regarded as expendable. There is, therefore, a need to produce reliable and efficient nozzles of simple design which can be easily manufactured in quantity. The nozzle must resist erosion during its short active life by choice of material and attention to design, as elaborate cooling devices cannot be introduced. Long-range missiles and high-altitude vehicles, such as meteorological sounding rockets, will also have expendable motors, but in these the highest thrust efficiency is demanded and greater complexity can be tolerated. The maximum range of a rocket is particularly sensitive to the structure/fuel-weight ratio and to the thrust efficiency. In practice, a well-designed nozzle may obtain a thrust efficiency equivalent to 96% of the thermal efficiency (as high as 60%) and even the least efficient can achieve 80%, so that small gains are of importance.

Nozzle-design theory is difficult to apply in practice. The usual design methods employed for wind-tunnel nozzles are not generally suitable, as they lead to excessively long shapes with high friction losses and also assume a set of initial conditions which are not generally found in a rocket-motor chamber. The best theoretical shapes are usually difficult to manufacture in quantity, because they require machining methods with a high degree of accuracy. Furthermore, a carefully designed and manufactured shape may erode and distort in operation, and there is evidence that the efficiency of these theoretical nozzles is sensitive to small errors in the profile. Thus, it has been the practice to design nozzles from empirical rules. This paper is concerned with establishing and rationalizing these rules, and with describing experimental work carried out over a number of years to find an efficient rocket-nozzle shape capable of simple manufacture. Experiments have been conducted with nozzles, generally with a ½-in. diameter throat, discharging air to atmosphere from reservoir pressures in the region of 900 psi. The thrust reaction and discharge rate have been measured, and the air flow within the nozzle and the issuing jet have been examined photographically. The apparatus used and the experimental techniques are fully described.

The investigations have determined the best design for simple nozzles of conical divergence and found these to have a thrust efficiency of some 96%. Suitable entry radius and divergence angle are recommended, and the effect of machining errors, of obstructions in and near the nozzle, and of operating at incorrect pressure ratios have all been considered experimentally.

In the course of this study, a number of unusual designs has been considered. These include annular nozzles, a variable-thrust nozzle, and a nozzle designed to be mounted at the front of a rocket. These unconventional nozzles all have some special feature which makes them particularly suitable for certain requirements.

The paper gives data on all the nozzle-design details investigated, and an attempt has been made to relate the observed thrust losses in conical nozzles to a simple theory. Throughout the research and in presenting the results, the requirements and interests of rocket-motor designers have been kept in view.

Heat Conduction through Insulating Supports in Very-Low-Temperature Equipment.

By R. P. MIKESELL and R. B. SCOTT. (From *Journal of Research of the National Bureau of Standards, U.S.A.*, Vol. 57, No. 6, December 1956, pp. 371–378, 15 illustrations.)

THE most efficient containers for storing and transporting low-temperature liquids, such as liquid oxygen, nitrogen, and hydrogen, are the vacuum-jacketed type known as Dewar vessels. In the past, several types of heat insulators have been used to bridge the vacuum space and furnish support for the inner container. For small laboratory-type Dewars, the design of insulators is not a major problem but, when designing a vacuum-insulated container of several hundred litres or more, sufficiently rugged to withstand the shocks and vibration of transportation by any type of carrier, the problem of insulating supports becomes serious.

In this paper, an apparatus is described which is used to measure the heat conduction through insulating supports of storage vessels for cryogenic liquids and presents the data obtained from the conduction measurements. Two types of supports were tested, i.e., (1) multiple-contact supports in the form of stacks of thin metallic plates or spirally wound strips, and (2) non-metallic spheres. The high thermal resistance of the multiple-contact supports arises from the numerous relatively poor contacts between the individual plates. Some special treatments of the plates were tried, and two of these (perforating and dusting), were found to be effective. Pyrex-glass spheres were also found to be excellent insulators, but are naturally not as rugged as a stack of metal plates.

The most important conclusions to be drawn from these results are that (1) the principle of multiple thermal contacts offers the designer of cryogenic equipment a simple rugged support which occupies small space, (2) the best assembly tested consisted of a stack of stainless-steel plates, 0·0008 in. thick, its thermal conduction when supporting a load of 1000 psi being only 2% of that of a solid conductor of the same dimensions, (3) the insulating value of such a support is enhanced by introducing a tiny amount of dust (MnO_2) between the plates, and (4) for best results the support should be designed for high loading per unit area.

The Corrosion of Lead Cable Sheath Under Cathodic Protection Conditions.

By W. H. BRUCKNER and R. M. WAINWRIGHT. (From *Corrosion, U.S.A.*, Vol. 13, No. 2, February 1957, pp. 73–78, 3 illustrations.)

THE question of whether lead surfaces can corrode under cathodic conditions has received considerable attention in recent years. Although the use of bare lead as a final covering for underground electric power and communication cables may be giving way to some kind of organic coating, there is still an enormous investment in bare lead-sheathed cable. It is recognized that lead does corrode when in soil or sea-water environments, and it would be highly desirable in most instances if this corrosion could be eliminated by the normal cathodic protection techniques. For this reason, any tendency of lead to corrode as a cathode is of major interest.

This paper describes laboratory tests conducted to determine whether or not corrosion of lead cable sheath can occur under cathodic protection conditions. Some of the runs were made in cells containing the aqueous

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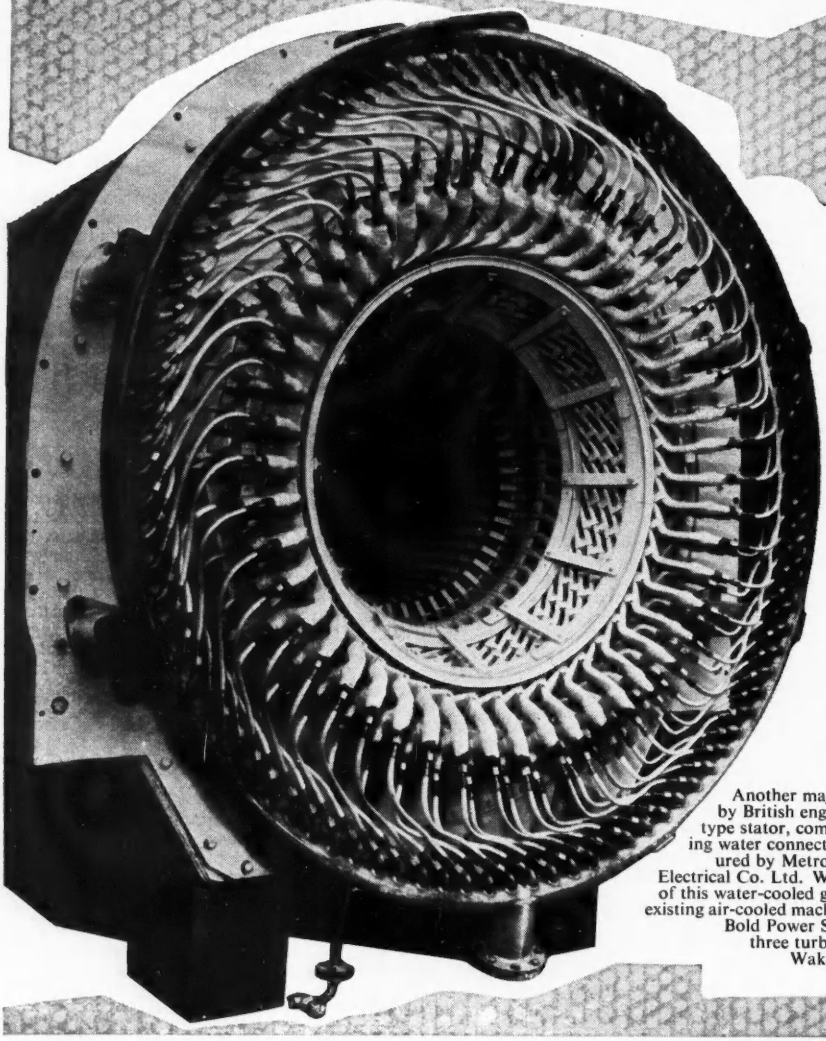

Another major achievement by British engineers—the new type stator, complete with cooling water connections, manufactured by Metropolitan-Vickers Electrical Co. Ltd. With the addition of this water-cooled generator to two existing air-cooled machines the C.E.A. Bold Power Station now has three turbines running on Wakefield-Dick oils.

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solutions of chlorides or chlorides and alkali, while other tests were made with the lead cable sheaths in contact with soil saturated with the various solutions. The results from all of these tests indicated that no corrosion of the cable sheath occurred in any of these environments, provided that the cathode was receiving protective current. It was found, however, that corrosion and pitting of lead cable sheath could occur when the protective current was interrupted.

It is considered that a system of lead-sheathed cable normally under cathodic protection may cease to have such protection as a result of the following service-connected factors:— (1) Formation of films and crusts of high resistance due to calcium oxide and magnesium oxide composition ; (2) seasonal removal of soil moisture or duct water ; (3) formation of a more highly conductive path of cathodic protection somewhere else in the system, as a result of heavy concentrations of de-icing salts ; and (4) daily and seasonal variations of stray currents.

Since the corrosive environment of lead cable sheath may be produced by the cathodic protection system itself, it will be appreciated that failure by corrosion in the absence of protective current will be more rapid with greater protective current density prior to cessation of protection. It would therefore appear that the term " cathodic corrosion " of lead cable sheath is a misnomer, and should be replaced in the literature by the more descriptive term " corrosion by cathodic products ".

Some reduction in the frequency of failure of lead-sheathed cable might be effected in the field by preventing the existence of excessive protective current. More precise inspection techniques could possibly determine when cathodic protection is absent as a consequence of the four factors mentioned.

Ultrasonic Attenuation in Steel and Cast Iron.

By K. KAMIGAKI. (From *The Science Reports of the Research Institutes, Tôhoku University, Japan*, Series " A ", Vol. 9, No. 1, February 1957, pp. 48–77, 13 illustrations.)

As described in this paper, ultrasonic attenuation in steel and cast iron of various textures was measured in the frequency range between 0·5 and 25 Mcs, utilizing pulsed longitudinal ultrasonic waves. For this purpose, several heat-treatment processes were applied successively to the same specimens, and at every stage of these treatments attenuation measurements were carried out. Chromium-molybdenum steel and carbon-steel specimens were normalized, quench-hardened, and tempered to make a troostite and sorbite structure, and the differences between attenuations at each stage were determined. For ascertaining the effect of austenite grain size on ultrasonic attenuation, the measurements were made in the case of the carbon steel with the specimens heat-treated above the transformation point. White cast iron was prepared, then tempered successively until spheroidal-graphite structures were reached, the attenuations being studied at every stage of tempering. Attenuation measurements were also made both for flake-graphite cast iron, and for spheroidal-graphite cast iron, and the effects of the shape and size of crystallized graphite on attenuation were studied.

The results of measurements may be summarized as follows :—

(1) In the case of steel, ultrasonic attenuation varies considerably with change in texture. In specimens with a pearlitic structure, a high attenuation value was observed, which rose steeply with frequency, so that the sound wave barely penetrated at frequencies higher than 10 Mcs. The predominant factor causing such high attenuation is the grain size of the austenitic phase. In quenched specimens of a martensitic structure, ultrasonic attenuation was fairly low at frequencies as high as 20 Mcs. In tempered specimens of a troostite or sorbite structure, lower attenuation values were obtained, these being considered to be caused by the elastically anisotropic character of each grain.

(2) In the case of cast iron, ultrasonic attenuation shows a complicated dependence on texture. The shape and size of crystallized graphite in cast iron have a remarkable effect on attenuation. White cast iron without crystallized graphite has a low attenuation value, while, in flake-graphite cast iron, very high attenuation values were observed, increasing sharply with frequency, this characteristic being determined directly by the size of the flake graphite. Cast iron of intermediate structure, including spheroidal-graphite cast iron, gave medium attenuation values.

(3) The predominant factor determining the magnitude of ultrasonic attenuation is the Rayleigh scattering of ultrasound by grains, the rate of energy loss by this scattering depending on the size and the elastically anisotropic character of the crystal grains.

Transmission Developments for Public-Service and Heavy Goods Vehicles.

By A. G. WILSON. (From a paper presented at a meeting, *The Institution of Mechanical Engineers, London*, March 12, 1957, 25 pages, 39 illustrations.)

TECHNICAL development in the field of transmission for vehicles powered by internal combustion engines has gained momentum with each succeeding decade since its inception at the turn of the century, but it is not generally realized how few basic ideas lie behind the complexities of new and up-to-date types and how closely the early pioneers anticipated the present developments.

This paper reviews the use, development, and history of the transmission system as applied to commercial vehicles and forecasts future trends. The point is made that the transmission system is to be considered in relation to the vehicle and the service to which the vehicle is put, rather than as an end in itself. The characteristics of the various systems at present available are therefore examined from the aspect of their effect on vehicle performance.

Since engines of similar characteristics are fitted into an ever-increasing range of vehicles of divergent types, features of the various types of transmission are analysed, to give a rough guide as to their suitability for differing service. For this purpose the modern types are grouped under a form of genealogical tree, in which an attempt is made to trace their ancestry. Commercial-vehicle usage is broken down into type of vehicle and the main markets into which they are sold, and the various contradictory features exhibited by the behaviour of the transmission system in the vehicles are listed with a suggestion of preference for the type of vehicle in that particular market. In addition, characteristics of different classes of vehicle are examined in relation to driver reaction and the type of control which would be most likely to appeal to drivers accustomed to them.

It is concluded that it is reasonable to deduce that the trends in heavy passenger and goods vehicles will be towards simpler gear changing and greater passenger comfort, even at the expense of higher first cost, observing that increased performance, high speeds, and less wear and tear on other parts of the vehicle can result from improvements in the transmission system. The reasons behind this are mainly economic, but technical progress towards improved facilities and conditions applies with the same force in the field of road transport as with other enterprises. Reduction of the physical effort of driving a commercial vehicle and the provision of greater safety for driver and passenger is a continuing process, which may be delayed, but cannot be prevented, by technical difficulties and higher costs.

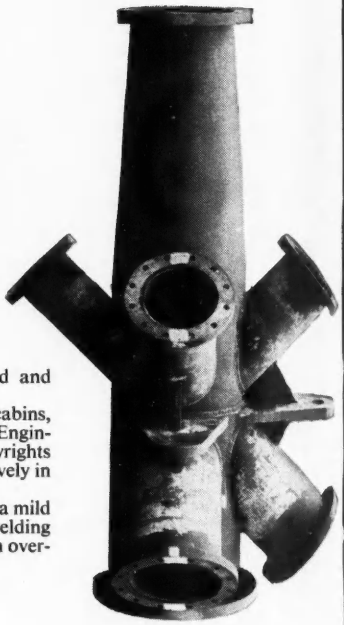
It is impossible to predict when a power unit will be available with all the characteristics needed to propel a vehicle with the requisite horsepower under all the various conditions likely to be encountered, but until then there is still definite scope for skill in the provision of transmission systems which will overcome the many problems at present existing.

COAL-BORING TOWER BUILT TO WITHSTAND 80 mph GALES

Lincoln D.C. Welding Machines and Multiweld O.V. Electrodes extensively used

The Coal-Boring Tower about to be towed to its working site. Its tubular structure lessens the resistance to wind and water pressure. Lincoln Multiweld Electrodes and D.C. Welding Machinery were extensively used on this project.

BELOW *This tubular structure is one of the main joints at the top of the four main members. These four joints have ten branches each and by tapering some of the members, the cluster of branch tubes and the heel pieces were welded with great precision.*



To assist in discovering under-seabed coal seams at Seafield Colliery, Firth of Forth, the National Coal Board are using a new boring system for the first time in Britain. Messrs. Maunsell, Posford and Pavry, consultants, recommended a stable drilling platform supported above high-water level by a 120-ft. tubular-steel tower floating on two independent 170-ft. pontoons. The hollow cruciform base, integral to the tower, is flooded and lowered to the sea bed by winches in the pontoons.

The tower can withstand 80 mph gales and its platform carries drilling rig, 25 cabins, a recreation room and a sea-water distillation unit. The Cleveland Bridge and Engineering Co. Ltd. were main contractors for both tower and pontoons, and Tubewrights Ltd. used Lincoln D.C. Welding Machines and Multiweld OV Electrodes extensively in fabricating the tower members.

Multiweld OV is extensively specified for large construction projects because as a mild steel electrode it has been specially designed for all-position high-speed fillet welding and its main characteristics are ease of flow, low spatter loss, easy slag control in overhead and vertical positions, and first-rate penetration.

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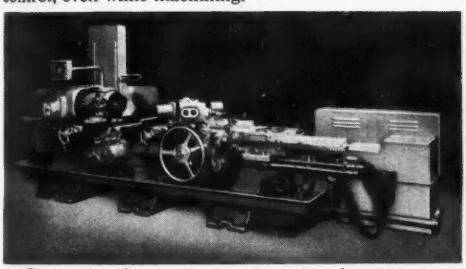
New Materials, Processes, and Equipment

UNIVERSAL TURRET LATHE FOR TWO-WAY HYDRAULIC TRACING

A new universal saddle-type lathe with a cross-sliding hexagon turret has been developed by the Jones & Lamson Machine Company, of Springfield, Vermont. By means of a selector switch, the lathe, which can be operated entirely as a standard machine, also provides the advantage of a two-way tracer capable of machining most inside and outside contours on chucked work.

Two-way hydraulic tracing from the cross-sliding hexagon turret is accomplished by the coordinated movement of its cross-slide and longitudinal slide. A single stylus controls the movements of both slides through hydraulic cylinders. Template control of the stylus provides tool travel in any direction (within its plane) towards the headstock, from a cut parallel to the spindle centre-line to a face 90 deg. to the centre-line. Since the facing can be both in or away from the centre, 180 deg. of tool control towards the headstock is provided. After completing one cut, the tool automatically relieves for rapid return. The tool can also be returned without relief and under feed for a second cut on the same contour. Uniform cutting feed is maintained on all work contours. The feed can also be varied by a single control, even while machining.



Conventional profiling with the hexagon-turret advance parallel to the spindle centre-line, or with the alternate advance at 90 deg. to the spindle centre-line, may be accomplished in the same chucking, using the same template. The machine is arranged to receive a flat template which is in the exact shape of the work to be reproduced. No modification or correction factor is required. All positions of the hexagon turret can be tracer-controlled, permitting the use of more than one tool tracing conventionally, or at 90 deg. to conventional, for profiling a difficult contour. Properly ground tools for rough and finish cuts can be used, and it is possible to profile the exterior, the interior, and the face on a workpiece in one chucking. This two-way tracer unit, in combination with the standard turret lathe, provides a capacity particularly suitable for difficult profiling on chucked work.

FLOOR-WARMING CABLES

Designed specifically to ensure long service and easy installation, the new range of floor-warming electric cables introduced by Aerialite Ltd., of Stalybridge, Cheshire, permits the full benefits to be derived from space heating by embedded floor-warming cables, with the maximum of safety, efficiency, and economy.

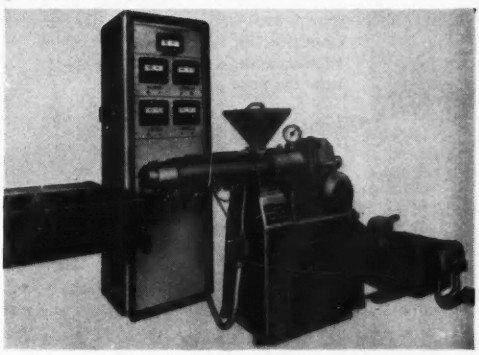
Each cable consists of a central conductor of high-resistance copper alloy (or, if required, of annealed copper) protected by a covering of tough, heat- and moisture-resistant enamel. Insulation comprises a polychloroprene compound which is a non-thermoplastic polymer, so that there is no relative movement between the conductor and the insulation. The overall sheath is made up of a special grade of P.V.C., selected for its heat-resistant, non-ageing characteristics. Both the insulation and the covering are fire-resistant and are virtually unaffected by moisture, oils, and solvents, and by the acids and alkalis associated with all types of concrete construction. Furthermore, because the cables are normally buried in concrete, oxidation effects are minimized.

Depending on diameter, the standard resistance of the copper-alloy conductors ranges between 0·028 and 0·191 ohm/ft, and of the annealed copper conductors between 0·0053 and 0·0080 ohm/ft. All these cables are designed in conformity with the standards generally accepted in Great Britain, i.e., a maximum emission of approximately 44 BTU/hr/ft^2 (13 W/ft^2), and a maximum floor temperature of about 72°F.

SCREW-TYPE EXTRUDERS FOR PLASTICS

A new range of screw-type extruders for all kinds of thermoplastics and for rubber has been developed by The General Engineering Co. (Radcliffe) Ltd., of Radcliffe, Lancs. The machine illustrated is the 1½-in. size, but 2½-, 3½-, and 4½-in. machines are also available. Simple construction, combined with adequate strength and power, ensures mechanical reliability, while the use of carefully designed screws and die-heads results in first-class extrusion quality at unusually high output rates.

The new machines are driven by an externally mounted variable-speed motor, rated for continuous duty and coupled to the extruder by a V-belt drive. Reduction gearing is of the worm-gear type and is kept separate from the thrust-bearing assembly, to prevent thrust loads from being transmitted back to the gears. An open chamber, between the feed and thrust boxes, precludes contamination of plastic materials by oil seepage, and also permits free fall-out of any plastic fines which may work back from the feed section. Provision is made for water-cooling the screw and feed box, and either air or water cooling can be provided on the barrel zones, when required, as an optional extra.



Direct electric heating of the barrel and die-head zones is provided, using cuff- and plate-type heaters. Temperature control is by proportioning instruments mounted in a cabinet, separate from the extruder. The barrel is in one piece, and small modern die-heads of many kinds are available for such duties as film and sheet production, tubes and miscellaneous sections, large pipes, cable insulation and sheathing, etc.

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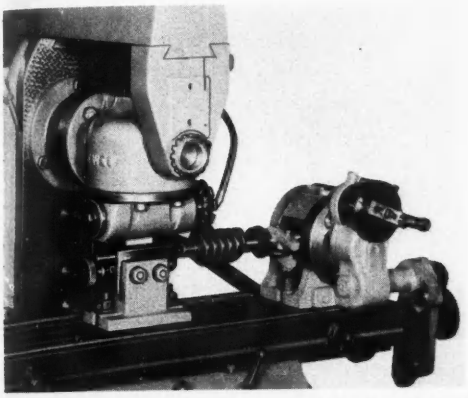
UNIVERSAL MILLING ATTACHMENT

Designed for all types of intricate milling operations, a new universal milling attachment, for use with their horizontal milling machine, has been introduced by The Greaves Machine Tool Division, J. A. Fay & Egan Company, of Cincinnati, Ohio. The new attachment is equipped with two graduated swivels, and the head may be rotated 360 deg. in each of two planes, to provide any desired cutter position. As a result, unusual flexibility is provided for operations not normally performed on a horizontal milling machine. In addition to complicated jig, fixture, and related work, the attachment may be used on a plain milling machine for milling spirals, when used in conjunction with a dividing head and lead attachment.

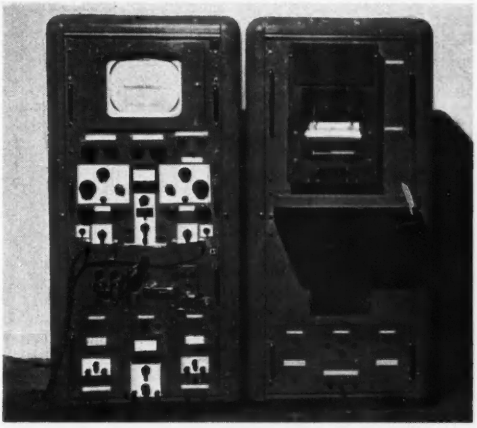


The unit mounts easily with three bolts to the machine-column face. The attachment is driven from the machine through helical bevel gears, with the drive shaft mounted in roller bearings. For ease in computing speeds for the material and cutter being used, the spindle speed of the unit is the same as that of the machine spindle.

ELECTRO-OCULOGRAPHIC EYE-MOVEMENT MEASURING APPARATUS

As a result of extensive research and development, the Psychological Research Laboratory of E.M.I. Electronics Ltd., of Hayes, Middlesex, has developed a practical complete system of eye-movement measuring apparatus for use in electro-oculography, i.e., the technique of electrically recording eye position and movements by detecting the resulting small d.c. potentials from the skin surface around the eye sockets. As a result, a definite step forward has been achieved in evolving methods of improving visual efficiency and of reducing or eliminating eyestrain and fatigue, particularly in the case of industrial and other workers engaged on exacting visual tasks.

In their new system, E.M.I. have succeeded in reducing very considerably the unwanted skin potentials which have so hampered investigators in this field during the past twenty years. Furthermore, they have found both the need and the way to make very pure electrodes, since as much zero drift can be derived from the electrodes themselves as from the skin. Basically, the operation of the system depends on the fact that the eye acts as a tiny battery, with a positive voltage at the cornea and a negative voltage at the retina. With the eye looking straight ahead, these voltages are in balance, so that an electrode placed on the skin at the side of the eye socket should register zero. As soon as the eye moves in any direction, however, the voltages are out of balance and can be picked up by the electrode, amplified by the equipment, and displayed on a cathode-ray tube or used to drive pen recorders, producing a continuous trace of



the horizontal and vertical movements. As these tiny eye voltages are of the order of 0·0005 microvolt, the extreme degree of sensitivity and accuracy which has had to be achieved by this measuring equipment will readily be appreciated.

As different subjects give different amounts of voltage per degree of eye movement, accuracy can be expressed only in statistical, and not absolute, terms. With an average subject tested in one-minute spells, it is now possible to record horizontal eye movements to better than $\pm$ 1·5 deg., and vertical movements to $\pm$ 2 deg. for 70% of the testing time. For shorter spells, e.g., 20 to 30 sec, the accuracy is better than $\pm$ 1 deg. for at least a similar percentage of the testing time. With selected subjects in one-minute spells, an accuracy of $\pm$ 1·5 deg. can be assured for more than 90% of the time, and better than $\pm$ 1 deg. for at least 70% of the time. By using a pen recorder and interpolating the zero level from the zero points at the beginning and end of the recording spells (one, two, or more minutes), it has been found possible to make a reliable analysis of the record to better than $\pm$ 1 deg. of eye movement for an entirely satisfactory percentage of the total recording time.

Part of the measuring apparatus is shown in the illustration. The pen recorders are on the right, while on the left can be seen the high-stability, high-gain d.c. amplifier specially developed by E.M.I., and the cathode-ray tube display, which was originated by Dr. N. H. Mackworth, of the M.R.C. Applied Psychology Unit, Cambridge.

MAGNETOSTRICTIVE LINEAR ACTUATOR FOR MICRO-FEED APPLICATIONS

Fulfilling the need for a powerful feed mechanism which is accurately controllable in the micro-inch range, the " Inchworm " motor, developed by Airborne Instruments Laboratory, Inc., of Mineola, N.Y., is a linear actuator providing controlled increments of relative motion between its two moving parts. Employing the principle of magnetostriction, the " Inchworm " may be considered as a logical successor to the lead-screw in all mechanisms requiring micro-inch dimensional control. Extremely rigid connection between the moving parts is effected, because there are no lead-screws, split-nuts, or bearings in the path of force transmission. In addition, as no lead-screw is employed, there is a positive improvement in the system spring-constant, owing to elimination of the torsional factor usually introduced by a lead-screw member. As a result, the system is not only extremely rigid, but is reversible and essentially free from backlash, enjoying the benefits of freedom from the wear-inducing lost motions encountered in oscillatory or vibrating systems. Furthermore, the " Inchworm "

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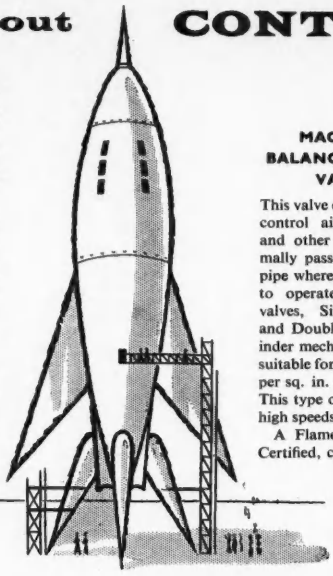

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actuator has brought the problem of stick-slip under control to the extent that the minimum increment of slide motion has been reduced by a factor of 15, compared with conventional feed mechanisms. In this connection, demonstration installations have consistently repeated to an accuracy of ± 5 micro-inches under machine load conditions requiring a break-away force of 300 lb and a sliding force of about 220 lb.

As indicated above, the new actuator obtains its motion by magnetostriction, i.e., the change in dimension occurring in certain materials when placed in a magnetic field (see page 149). Thus, for instance, a rod of properly heat-treated nickel, 10 in. in length, can produce a change in length of 300 micro-inches when saturated in a magnetic field under ideal conditions. Naturally, the external forces applied to the " Inchworm " actuator will (within the elastic limits of the materials involved) directly influence step size. Consequently, while a typical actuator, with a 4-in. field of magnetic influence applied to the nickel armature, has a theoretical step size of 120 micro-inches, in practice the step size is approximately 60 micro-inches under a 300-lb external load. Hence, the spring constant at 300 lb may be said to absorb 50% of each step in this instance.

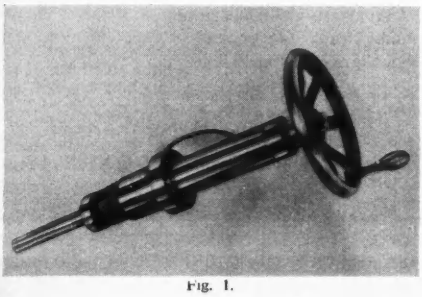

Fig. 1.

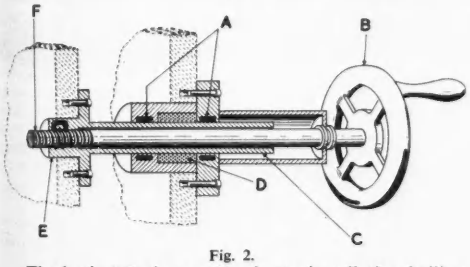

Fig. 2.

The basic " Inchworm " prior to installation is illustrated in Fig. 1, while a cross-section is shown in Fig. 2, from which it will be seen that the magnetostrictive armature *C* is supported in a pair of co-axial clamps *A*, mounted in a rigid machine element and spaced to provide the desired maximum step size. Motion is produced by magnetically shrinking the armature *C* while the right-hand clamp is locked and the left-hand clamp unlocked, thus moving the left end of the armature to the right. The left-hand clamp is then locked, and the right-hand clamp unlocked, after which the armature is expanded to its normal dimension by removing the magnetic field produced by the coil *D*. At this stage, then, the right end of the armature has moved to the right, and a net motion between the two members has taken place.

The clamps *A* incorporate a design which produces very little shift between the movable and rigid members when they are clamped and unclamped. The clamps consist of a fluid-filled cylindrical groove covered with a membrane which contacts the armature *C*, and, when fluid under pressure is applied to the membrane, it deforms and presses against the armature, forming a friction lock. Removal of the pressure permits the armature to slide freely within the clamp.

Locking and unlocking of the clamps may be controlled by varying the fluid pressure, the sequencing operation being controlled automatically. For manual control, a single three-position switch, arranged for both forward and reverse actuation, is provided, together with a step-size control in the form of a variable transformer. For automatic control, a gauge and statistical programming device provide the step signal, step size being controlled by variation of the amount of current flowing through the magnetizing coil *D*.

Since most machine-tool operations require a coarse or set-up feed facility, in addition to a vernier or control-feed facility, the " Inchworm " actuator is fitted with co-axial coarse-feed components, as shown in Fig. 2. In this type of assembly, the coarse-feed wheel *B* is used to set up the machine elements manually by means of ball-worm and ball-nut components mounted on the movable machine element. In practice, the " Inchworm " vernier feed overrides the ball worm *F* and the ball nut *E*, permitting the coarse feed to be used only when both clamps of the fine-feed mechanism are relaxed.

In semi-automatic operation, the operator of a centreless grinder may, for instance, require a 50-micro-inch wheel-position change to compensate for wheel-size error. By adjusting the step-size control dial to the proper mark and deflecting the forward-reverse switch for the desired feed position, the required feed is obtained quickly, accurately, and dependably, without the danger of overshooting so often experienced when converting rotary motion into linear motion with a lead-screw in-feed mechanism. Insofar as automatic systems are concerned, the ease and accuracy of positioning possible with the " Inchworm " actuator make the set-up and operation of centreless grinders extremely simple, even for relatively inexperienced operators. Using an electronic gauge to monitor the regulator wheel-slide position during set-up, the slide may be rapidly and repeatably positioned to within 5 micro-inches of the desired position. In addition, the anti-backlash feature permits very accurate control of the bi-directional motion required for continuous repositioning of the regulator wheel-slide in providing uniform size output, despite gradual wheel build-up or breakdown and variation of input-size scatter.

" FREEZING " OF THIN METAL HONEYCOMBS FOR EASIER MACHINING

An interesting method of facilitating the rapid and economical production of the extremely thin and fragile metal honeycombs used to reinforce vital aircraft and missile components has been developed in the U.S.A. by Northrop Aircraft Inc. and is now in general use in their plant.

Previous methods of cutting this material, which, by virtue of its honeycomb structure, provides great strength in relation to its weight, were negatived by its inevitable tendency to crush under the impact of the cutter. Now, however, all difficulties in this connection have been circumvented by the simple expedient of freezing the honeycomb in ice. In practice, water is poured into the cells of the honeycomb and, by means of a refrigeration unit coupled to a standard milling machine, freezes in a few minutes, stiffening the material so that it can readily be machined without fear of crushing. The rotating tool used for cutting is of the " bacon-slicer " type, operating at up to 35,000 rpm, and is fed across the work manually, cutting the honeycomb smoothly and rapidly. Cutters vary in size from 6 in. to ¾ in. in diameter, depending on the depth of cut required.

A vacuum chuck is used to hold the work, a piece of phenolic or aluminium sheet being cemented to the bottom of the honeycomb, so that the chuck can hold down both the sheet and the part. The sheet may then

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remain as a permanent skin or can be removed, according to requirements. A vacuum-pump, linked to an accumulator tank, prevents surge in the vacuum-chuck system.

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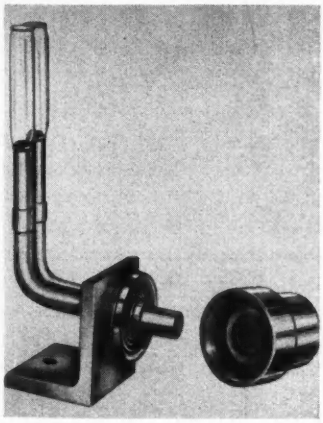
ULTRA-HIGH-VACUUM VALVE

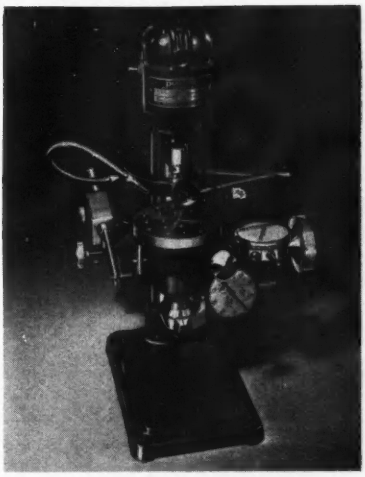
A new laboratory ultra-high-vacuum valve for handling pressures of 10^{-10} mm Hg and lower is now available from the Rochester Division of the Consolidated Electrodynamics Corporation, of Pasadena, Cal. Designated the VAU-25, this valve is used in ultra-high-vacuum systems where non-contamination and the ability to be baked out are of highest importance. The effective conductance of the valve for nitrogen is only 10^{-11} litre per second. The VAU-25 valve will seal a vacuum system from a diffusion pump without the contamination and inconvenience of a glass seal-off. It is also useful in handling very pure gases, such as in the study of breakdown characteristics of noble gases, where the system must not only be reduced to ultra-high vacuum, but gases must also be transferred from one portion of the system to another without introducing impurities. In such cases, greased stopcocks or mercury cut-offs could introduce vapours with pressures far greater than the ultimate pressure attainable.

The valve consists of the valve assembly, which becomes part of the vacuum system, and the driver mechanism, which opens and closes the valve. Because they are separate, the valve can be baked out at high temperatures (up to 450°C), without also baking the driver threads and lubricant. The valve assembly comprises a Monel cup sealed off by a flexible Monel diaphragm which can move 0·1 in. Opening into the Monel cup are two ¼-in. copper vacuum leads. Opposite these leads and attached to the inside of the diaphragm is a Monel nosepiece with a highly polished 45-deg. conical surface. The driver mechanism contains a differential screw and is rotated by a torque wrench. The differential screw exerts a force of 1 to 2 tons on the Monel nose of the valve assembly, forcing it against one of the ¼-in. orifices in the cup and closing off the vacuum lead.



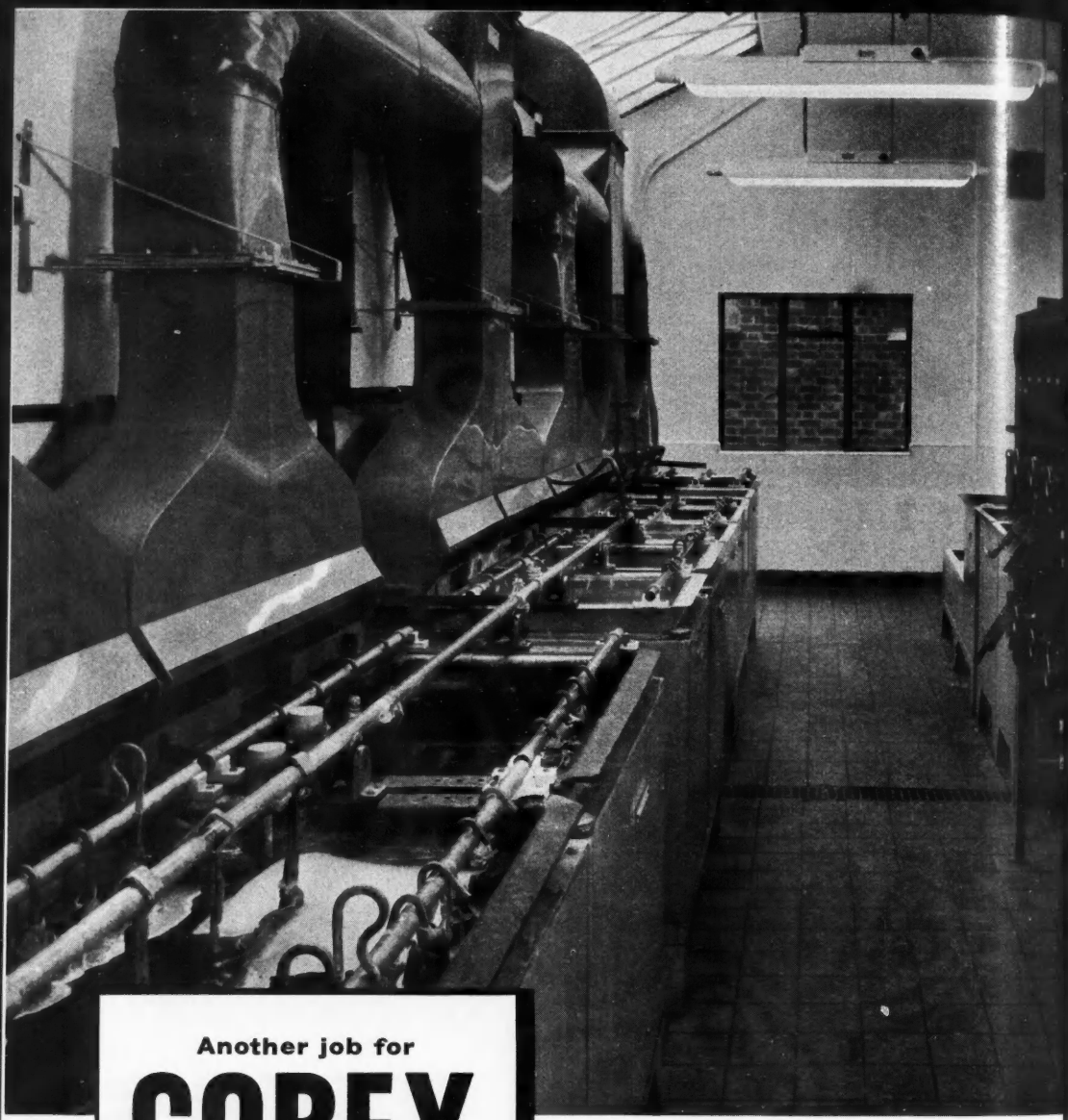
PRECISION MICRO-DRILL WITH CENTERING MAGNETIC CHUCK

Accurate chucking and the maintenance of drill concentricity are major obstacles to the successful drilling of holes in the micro-drill range but, with the new micro-drill announced by The Dunmore Company, of Racine, Wisconsin, difficult small-hole drilling operations down to 0·001 in. have been simplified and speeded up. Accurately indicated two-way table travel permits quick location and exact spacings between holes simply by turning conveniently located knobs. Two large dials indicate precise table movement, with a travel range of 1⅛ in. laterally and from front to back. The table is drilled and tapped for holding fixtures.



An accurate and instant-centering magnetic chuck provides fast change-over on different drill sizes without stopping the machine. Once the drill has been inserted into the collet, it can quickly be trued with a pencil point, the result being an accurate, concentric drill rotation essential for preventing drill breakage. A highly sensitive table feed is accomplished through the use of an adjustable counterbalance which offsets the weight of the table and fixtures. A calibrated micrometer depth control provides a positive stop. Tolerances of 0·0005 in. can be maintained on blind holes. To provide the speed of up to 2000 rpm required for micro-drilling, an adjustable auto-transformer for voltage control is used, the top motor speed being 17,000 rpm. A ten-power viewer and spot pinpoint drilling operations, providing full, clear vision on the smallest holes.

When drills of different diameters are used intermittently, considerable time can be saved by using extra collets for holding the various sizes of drills and quickly interchanging them, as required. This type of chucking device provides a quick method of changing drills and a simple way of truing, which can be effected while the machine is in operation.



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NEWS OF THE MONTH

PERSONAL

Mr. Derek Foster has been appointed Sales Liaison Manager to The Owen Organization, and will operate from Owen House, 11 Greenfield Crescent, Birmingham 15, telephone number Edgbaston 4566.

Mr. M. F. Dowding, Engineering Sales Manager of Davy and United Engineering Company Limited, has been appointed a Director of Davy-United's Middlesbrough subsidiary, Davy and United Roll Foundry Limited, manufacturers of rolls and steel castings.

Mr. B. Butcher, Organizing Superintendent and Labour Manager of Samuel Fox and Company, Limited, a subsidiary of The United Steel Companies, Limited, has been appointed a director of Samuel Fox from 21st February.

Mr. H. Smith, who has been Joint Managing Director of the Dyestuffs Division of Imperial Chemical Industries Ltd. since 1955, has been appointed Joint Managing Director of the General Chemicals Division. **Dr. C. R. Mavin,** previously Dyestuffs Division Production Director, has succeeded Mr. Smith as Joint Managing Director of that Division. **Mr. R. S. Wright,** previously Production Director of the Pharmaceuticals Division, has been appointed to the Dyestuffs Division Board as Production Director. **Mr. J. Grange Moore,** who has been a production manager of the Metals Division since 1955, has been appointed Works and Personnel Director of the Wilton Council.

The appointment is announced—to take effect as from 1st April—of **Mr. J. I. Bernard, B.Sc.Tech. (Hons)., M.I.E.E., M.I.H.V.E.,** as the new Director and Secretary of the British Electrical Development Association in succession to **Mr. V. W. Dale, Companion, I.E.E.** Mr. Dale, who intimated his impending resignation some months ago on account of health considerations, will continue to be associated with E.D.A., until 1st July, in order to conclude existing commitments. Mr. Bernard joined the staff of E.D.A. in 1928 and has been its Chief Technical Officer since 1936.

Sheepbridge Engineering Ltd. announce that **Mr. E. A. Cooke, M.I.Mech.E.** has been appointed Scientific Engineer to the Sheepbridge Group. Mr. Cooke was Chief Development Engineer at W. T. Avery Ltd. immediately before joining the Sheepbridge Engineering Group and before that head of Mechanical Development at Taylor Taylor & Hobson Ltd. Mr. Cooke's present appointment is primarily concerned with the development of specialized equipment and techniques.

Mr. R. Turner, Chief Engineer of Powell Duffryn Technical Services Ltd., has been appointed to the Board of that company in place of **Mr. G. W. Alexander,** Chief Engineer of Powell Duffryn Ltd.

Mr. A. R. W. Low, the former Minister of State, Board of Trade, has been appointed a Director of the General Electric Company Ltd.

Coventry Gauge & Tool Co. Ltd. announce that Mr. E. **Crabtree** has been appointed Technical Sales Representative for the Midlands Area.

ALBERT EDWARD GRIFFITHS

It is with deep regret we announce that **Albert Edward Griffiths,** Founder and Chairman of A. E. Griffiths (Smethwick) Ltd., passed away at his home, Milverton Grange, West Park Road, Smethwick, on Friday, 1st March, 1957.

BUSINESS NEWS

W. E. Sykes Ltd., Manor Works, Staines, Middx., the well-known specialists in gear cutting machines and tools, have produced a thirty-six frame filmstrip which is available on loan to lecturers at Technical Colleges and similar institutions, Works Study Groups, etc. The strip is of a double-frame size on standard, perforated 35 mm film, and is accompanied by a booklet of notes intended to assist in the preparation of a lecture on gear cutting. It must be emphasized that the booklet does not contain material for a complete lecture, but is only intended as a guide and a connecting link between the filmstrip and the lecture.

On the 1st of April 1957, **Hepworth & Grandage Ltd.** celebrated its Golden Jubilee—50 years of growth and progress. In 1907 E. Hepworth & Sons consisted of one small machine shop and a handful of men. Today Hepworth & Grandage Ltd., makers of Internal Combustion and Gas Turbine engine components, employ over 4,000 people. Their products are fitted in many British vehicles and engines, and their replacement engine components are famous in this country and throughout the world.

Due to continued expansion, the Middlesbrough Branch Office of **Honeywell-Brown Ltd.,** has moved to larger premises. All enquiries concerning industrial instrumentation, heating and air conditioning controls, and precision switches, should be addressed to:—Honeywell-Brown Ltd., 52–60 Fletcher Street, Middlesbrough. Telephone number: Middlesbrough 44221/3. Telegrams: Minnreg, Middlesbrough.

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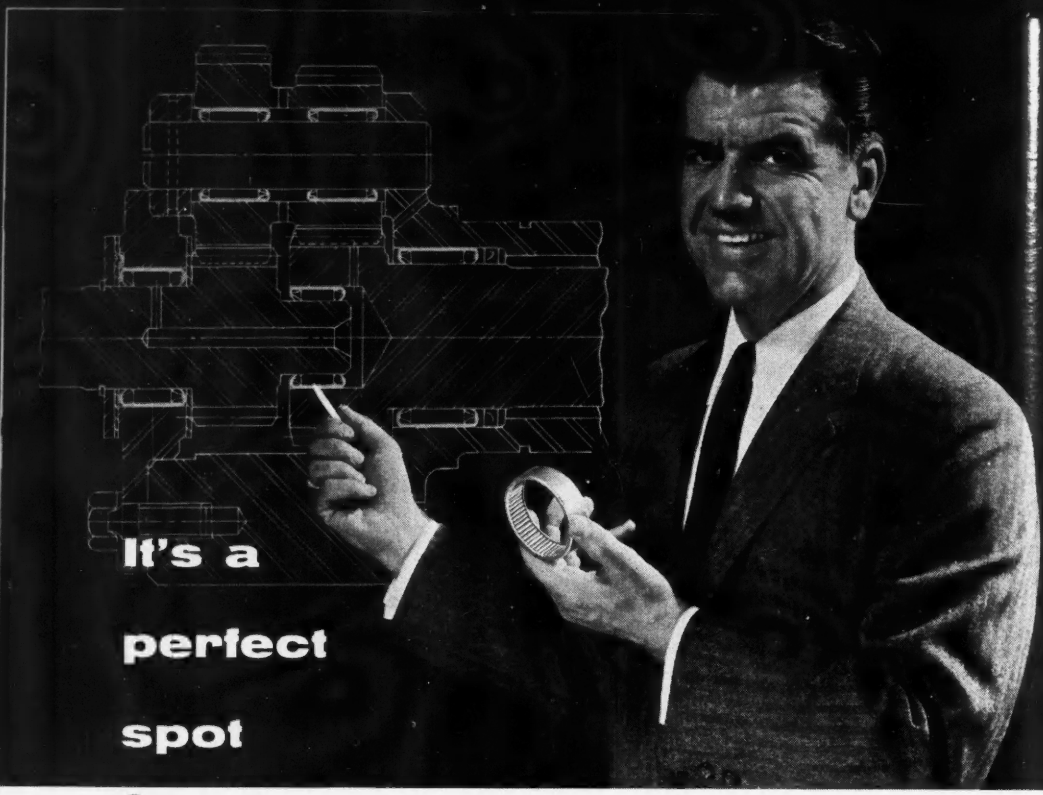

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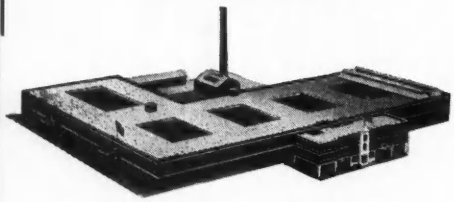

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BUSINESS NEWS

A modern five-storey building in the centre of Liverpool has been purchased by **Automatic Telephone & Electric Company Ltd.**, for use in connection with the Company's continued expansion in the electronic field. The building, 39–87 Cheapside, has a total floor area of 55,000 square feet, and a site of more than 3,000 square yards. It was formerly owned by Norman & Burgess Ltd., stationers and printers. In addition to the main A.T.E. factory at Strowger Works, Edge Lane, Liverpool, the Company already operates several satellite factories in the area, producing a wide range of telecommunications equipment.

Crompton Parkinson Ltd. announce that their Darlington branch has been moved into new premises and that the address is now at Crompton Parkinson Ltd., King Street, Darlington. Telephone No. Darlington 5285/6.

A programme of expansion in overseas markets is evident in the announcements which have been issued over the past few months by the **Consolidated Pneumatic Tool Company Limited** of London and Fraserburgh. This receives further emphasis with a statement just released advising the formation of a new company under the title of the **Consolidated Pneumatic Tool Co. (India) Private Ltd.** The new company which came into being on February 28th, 1957, has a Directorate comprising Mr. G. J. Coffey, Mr. Norman Readman and Mr. V. Nimbkar (Managing). It will operate in Bombay where new factory premises will be constructed.

Due to newly organized production extensions, William Jessop and Sons Ltd., Sheffield, manufacturers of high-speed steel toolbits, are now providing an " off the shelf " service of a range of over 90,000 toolbits, available for immediate delivery in 72 different standard sizes, The company issues a monthly stock list on request.

Owing to greatly increased demand for Jenolite products in the industrial centres of the north of England, the Jenolite Group of Companies have launched a new organization to distribute and service their products and processes in these areas. This Company, **Jenolite Industries (Northern) Limited,** with its headquarters in Huddersfield, are now handling all of the products manufactured by Jenolite Limited, the Jenolizing Company Limited and Jenograph Limited. These include metal degreasers, paint strippers, descaling and derusting processes, phosphating processes, aluminium treatments, de-watering oils and stains, and natural graphite lubricants.

British Insulated Callender's Cables Limited, announce that the alterations and extensions to their Edinburgh Branch Office at 80–82 Hanover Street, have now been completed. The offices have been removed to the first floor, so that the ground floor may now be used for storage, thus doubling the store capacity. This will enable a larger and more comprehensive range of cables and accessories to be readily available from stock.

Midland Industries Limited, Manufacturers of the MIL range of Steam Traps, Reducing Valves, announce the opening of their Scottish Office at 69 St. Vincent Street, Glasgow C.2.—Tel. No. City 7202. This office will be in operation as from April 1st, 1957 and will carry stocks of their standard equipment in order to give immediate service and supply to their many customers in Scotland. Mr. E. H. Fieke, A.M.I.Plant.E., formerly Senior Technical/Representative for the North/East, has been appointed Manager for Scotland. Mr. A. Paterson, formerly of Messrs. A. C. & W. Elliot, has joined the Company and has been appointed Technical/Representative for Scotland. Mr. A. Tye has succeeded Mr. Fieke as Technical/Representative of the North/Eastern area.

EXHIBITIONS AND MEETINGS

Owing to the fact that the Annual General Meeting of **The Institute of Mechanical Engineers** has had to be postponed from Friday, 22nd March to Friday, 26th April, it has been found necessary to rearrange the meeting programme for the remainder of this session. The changes are as follows:—1. Annual General Meeting postponed from Friday, 22nd March to Friday, 26th April. 2. General Meeting postponed from Friday, 12th April to Friday, 3rd May, at 6 p.m. 3. General Meeting postponed from Friday, 26th April to Friday, 17th May, at 6 p.m.

The Department of Management Studies of **The Polytechnic,** Regent Street, London, W.1, is offering a new day-time course of ten sessions on " Effective Speaking " for senior executives. The course will begin on Monday afternoon, April 29th. Each weekly session will be from 2–5 p.m.

A new evening course of ten lectures on " Communication in Management " for management executives begins on Thursday, May 2nd.

The popular day-time course of ten sessions on " The Interviewing Techniques of Management " for managers and supervisors is being repeated. The course will begin on Friday, May 3rd. Each weekly session will be from 2–5 p.m.

A new day-time course of six sessions on " Training in Manual Skills " for works managers, personnel and training officers, and senior foremen begins on Friday, May 3rd. Each weekly session will be from 2–5 p.m.

A complete report of the **Aircraft Production Conference** held at the University of Southampton on 8th/9th January, 1957, will shortly be available in book form. The price per copy (including postage) for Members of the Institution is 7*s.* 6*d.* and for non-members 10*s.*

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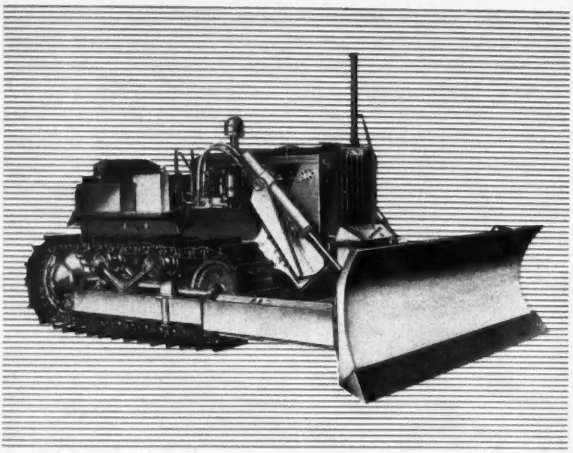

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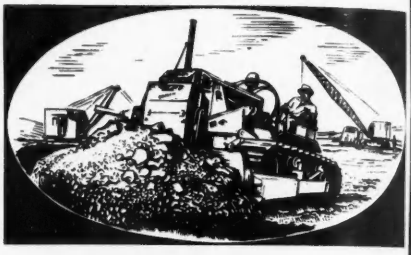

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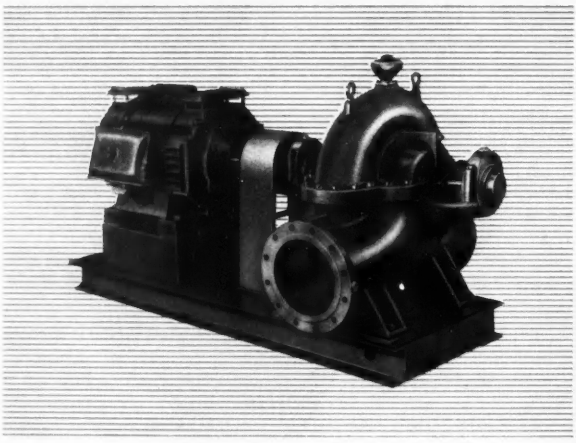

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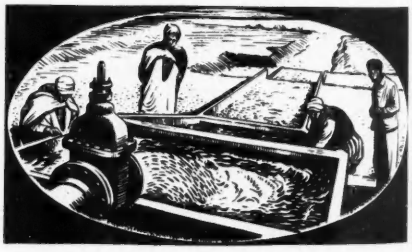

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The English Electric Company Limited announces that, following adjudication of various competing designs, it has received an order from the Admiralty for the design and manufacture of the main propelling steam turbines and condensers for Britain's first nuclear powered submarine. This order covers the prototype machinery, which will be tested in a land-based hull, and also the turbines to be installed in the first sea-going submarine.

New types of self-adhesive tape developed by **Industrial Tapes Ltd.** are " Double-Sided ", i.e., both sides have the self-adhesive " stick-at-a-touch " character. Double-Sided Speedfix tapes are rolled up with an interleaving uncoated strip which is peeled off and discarded in use. The Double-Sided tapes are being successfully used for the holding of non-ferrous metals and plastics on the magnetic chuck or steel-block of horizontal grinders. For holding of small ferrous metal parts on magnetic chuck, where metal part is too small to be held in position by magnetic field. Holding of ferrous metal parts, on magnetic chuck where the magnetic field would distort the shape of the unit. Holding of flat plate, washers, etc., on face-plates.

Simms Motor Units announce that the registered trade name for their quick-action inertia starter is " TenSec "—an adaptation of the words ten seconds which is the maximum time taken to operate the starter. The TenSec manual starter now in production for deliveries later this year, was first announced at a motor trade and press luncheon in December, 1956. The Simms TenSec starter applies with particular advantage under conditions where battery maintenance is difficult and costly, as in all tropical countries. It has already been widely ordered in this country and overseas.

A new range of general purpose squirrel cage induction motors is being manufactured by **Hoover (Electric Motors) Limited** at Cambuslang, Lanarkshire. The dimensions conform to British Standard 2048/1953 and the electrical performance to B.S. 170/1939. The new Hoover Mark IV Motor, which is cylindrical, has been designed to conserve material and reduce weight and size and covers a useful horsepower range from approximately $\frac{1}{16}$ to $\frac{1}{4}$ (4 pole 50 cycles). The diameter has been reduced to slightly less than $5\frac{3}{4}$ in. whilst the weight of the $\frac{1}{4}$ H.P. motor, for example, is only 19 lb. The new Hoover Mark IV Motor does not, however, supersede the widely known " square " Mark II Motor, which will continue to be manufactured.

The **United States Steel Corporation** announces that it has enlarged the group of prominent steel producers who are licensed to manufacture and sell its Cor-Ten steel, best known and most widely available of the corrosion-resistant, high-strength, low-alloy steels, by licensing the South African Iron and Steel Corporation (ISCOR) of Pretoria, Union of South Africa, and de Wendel & Cie., S.A. of Paris, France. The addition of these two companies raises to fifteen the number of steel producers licensed by United States Steel to use the trade mark.

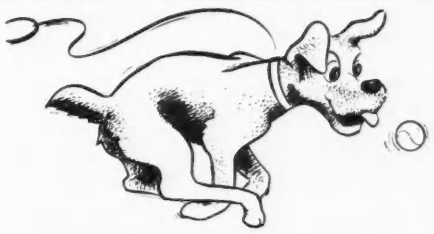
Teddington Industrial Equipment Ltd., have reduced the price of their OV " Engine Watcher " from £27 10*s*. to £25 as a result of improved methods and increased production. The demand for this automatic engine protection unit necessitated increased production at the Sunbury-on-Thames Works. Reorganization of manufacture resulted in a saving in production costs.

Standard Telephones and Cables Limited have been making transformers for more than fifty years, but this production has been used almost exclusively in the wide range of telecommunication and power equipments manufactured by the Company. The design and manufacturing experience gained over this long period is now being used for the production of standard ranges of power transformers for general sale.

An ingenious piece of apparatus incorporating spring balances loaned by **Geo. Salter & Co. Ltd.**, of West Bromwich, is in use at the **Industrial Safety Training Centre,** Birmingham, to demonstrate safety principles while hoisting crane loads. Beneath a 10 cwt capacity crane weigher are a pair of No. 235 200 lb circular balances which take the place of the normal rope or chain slings. The angles of the balances, at the point of attachment to the load, can be varied, and the dial readings on the balances demonstrate that the load imposed on the " slings " increases as the angle widens. Thus, with too wide an angle, a sling may be subjected to a loading beyond its safety limit.

The Engineer Research and Development Laboratories of the United States Army have placed a contract with **D. Napier and Son Ltd.**, Acton, London, W.3, for a compressor set, consisting of an 18-cylinder " Deltic " diesel engine and two Howden rotary positive displacement two-stage compressors, for use with a mobile liquid oxygen plant. The set, which is completely self-contained, and weighs 43,000 lb can be transported by road or by air; when mounted on a road trailer it is approximately 8 ft 10 in. × 11 ft × 35 ft overall, and has a gross weight of 50,000 lb. It will supply up to 30,000 lb of dry air per hour at 100 lb/sq in.

One of the main limiting factors to the rapidly increasing production in this country of glass fibre reinforced plastics is the restricted supply of suitable fillers. To help ease this shortage, **Jo'n & E. Sturge Ltd.** are extending certain production facilities at their Lifford Chemical Works, Birmingham, to increase by 300% their output of Calofil, a special grade of precipitated calcium carbonate used widely as a filler for polyesters. Calofil is a white, low oil absorption, non-abrasive chemical which is claimed to be the only specially manufactured inert filler for reinforced plastics produced in this country.



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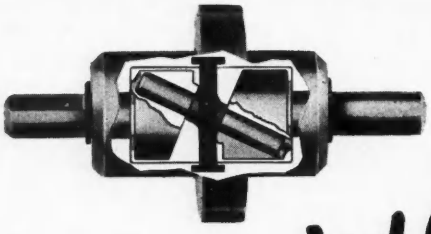

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Technical Session on Electrical Equipment of Machine Tools

The discussions on electrical equipment of machine tools were held under the chairmanship of Sir Stanley Rawson (Vice-chairman, John Brown & Co., Ltd.). In the absence of any other standards within the Commonwealth on electrical equipment for machine tools, the delegates agreed that the recently published British Standard 2771, Electrical Equipment for Machine Tools, should be taken as a basis for discussion It was realized that it would not be possible at the present time to draw up a general Commonwealth standard acceptable to all members, but B.S. 2771 was examined in detail and certain of its clauses amended so as to make them generally acceptable throughout the Commonwealth. A new draft incorporating these amendments is being circulated by B.S.I. to the other standards organizations and it is hoped that these recommendations will be followed in the preparation of all Commonwealth standards. In this way equipment produced in any of the Commonwealth countries would be acceptable throughout the Commonwealth. The problem of tropical conditions was discussed in detail. It was not considered possible to lay down in precise terms the requirements which should be satisfied by equipment destined for use in the tropics. It was, however, agreed that a note should be added to the document (based on B.S. 2771) warning manufacturers that electrical equipment to be fitted in machine tools intended for use in tropical countries should be such that it would give adequate protection under the conditions likely to be experienced.

Technical Session on Steel

The discussions on steel were held under the joint chairmanship of Sir J. J. Ghandy (Director-in-Charge, Tata Iron and Steel Co., Ltd.) and Mr. E. W. Senior (Commercial Director, British Iron and Steel Federation). Discussions were held on the following: 1. General problems of standards for the use of steel and their implementation. 2. Redesign of hot rolled steel sections. 3. High tensile structural steel. 4. Welding of steel structures. 5. The use of steel in structures. 6. Standardization as applied to tubular steel structures. The Commonwealth countries have already achieved a large measure of co-ordination of steel standards and the principal purpose of the Delhi meetings was, therefore, to achieve closer alignment. India had prepared a paper emphasizing the important role of standards and their implementation in promoting the more efficient use of steel, this being particularly important to India because of that country's shortage of steel and the large programme of expansion which is envisaged. The Australian delegation stated that, in general, they followed British Standards, and although there were only two steel plants in the country attempts were being made to rationalize the number of types of steels produced.

A paper prepared by India on The Redesign of Hot Rolled Steel Sections emphasized that this was an important part of her steel economy programme. Indian standards for beams, channels and angles had been finalized and the draft standards for bulb angles and tee bars were ready for circulation. The United Kingdom delegation felt that one of the dangers resulting from introducing a further series of sections was that a larger variety than hitherto would be rolled. It was feared that the adoption of new standards in India would be a hindrance to the co-ordination of standards for structural sections within the Commonwealth.

A paper presented by the United Kingdom on High-Tensile Structural Steel pointed out the advantages where weight reduction is important, as in ships, bridges, jib booms and wagons. The availability of high-tensile steel in the United Kingdom presented a difficult problem and the question of obtaining these steels from stock had not yet been satisfactorily settled, but the position could be considerably helped by keeping the number of sections to a minimum.

Another U.K. contribution on Welding of Steel Structures brought out the sound methods adopted in Britain for preparing welding standards. It was noted that there was considerable need for closer Commonwealth co-ordination on standards for terminology, symbols, classification of electrodes, weld preparation, approval tests for welders, inspection of welds and welding equipment. The Indian delegation stated that, despite the large amount of work which had been done in India, there was considerable scope for improving her welding techniques to bring them into line with more industrially-advanced countries and she was therefore anxious to take advantage of the experience of other countries.

A paper prepared on The Use of Steel in Structures by the United Kingdom was based on B.S. 153, Girder Bridges, and B.S. 449, Use of Structural Steel in Building, which it was thought could serve as a useful basis of design throughout the Commonwealth. From the discussions it became apparent that much closer co-ordination in this type of work was necessary to ensure that the various national standards would be acceptable throughout the Commonwealth.

The Indian Standards Institution is finalizing a draft for the use of tubes in general building construction; it is in line in most respects with the provision of the revised draft of B.S. 449, except for the provisions relating to the strut formula where, for a wide range of $\frac{L}{R}$, the permissible stresses in the Indian Standard will be appreciably higher than those permitted in Britain.

Inch-Metric Problems

One of the most important subjects discussed was the effect of India's adoption of the metric system on intra-Commonwealth trade. This change-over poses special problems in the alignment of Commonwealth standards.

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Details of other surveys will be announced later.

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On the Indian side it was made clear that the formulation of standards in the metric system was a matter of great urgency. India is anxious, however, to adopt standards which are as far as possible in line with those of the inch-pound system, and, in particular, will choose its international "preferred" numbers (on which are based ranges of sizes), as closely as possible in agreement with standards on the inch system. There is to be an urgent review of India's dimensional standards to decide which can be adapted by simple conversion from the inch to the metric values, those where inch and metric dimensions can be unified from the point of view of interchangeability and performance and those where they cannot be so unified. Commonwealth countries will also co-operate in trying to achieve the greatest possible reconciliation in ISO (International Organization for Standardization) and I.E.C. (International Electro-technical Commission) of the inch and metric systems on the lines of proposals already made to ISO by the U.K. A committee will be set up in Britain to work in close collaboration with an Indian committee to arrive first at metric equivalents of existing inch dimensions for all cables, indicating certain preferred sizes, and then to agree on the definite metric sizes.

"A.B.C." Progress

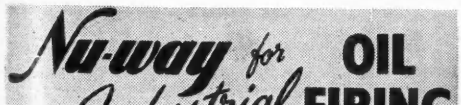
A review was made of the progress achieved by the "A.B.C." countries (America, Britain, Canada) on fundamental engineering standards, and of the association of other Commonwealth countries with this work.

Public Authority Purchasing

Attention was focused on the need for greater co-ordination of inspection procedures between the military Services and for avoiding duplication of inspection for industrial and Services' requirements. The conference recommended particularly that public purchasing organizations should take advantage of the guarantees afforded by certification marking schemes run by standards organizations. This would enable public authorities, including the Services, to dispense to a large extent with their own inspection procedures for some items.

New Work

One of the important subjects discussed was the further items of industrial equipment which should be subjected to detailed technical consideration in the same way as cables, steel and so on at the 1957 conference. It was agreed that the next subjects to be tackled should be standards for **air receivers** and **cranes**, where divergent statutory regulations in the Commonwealth countries complicate the exporter's and importer's problem and where co-ordination of testing procedures and design criteria is particularly needed. In view of the work going on in ISO it is not proposed to initiate any special Commonwealth discussions on **boilers**, though a review will be made of the extent to which Commonwealth countries will be able to follow any ISO recommendations. **Chemical pressure vessels** have been noted as a possible subject for discussion later. Following discussion on the desirability of bringing **terminology** in the Commonwealth and the United States more into line, it was agreed that a special study should be made by B.S.I. in co-operation with other Commonwealth standards organizations, of the terminology used in (a) welding and (b) building design and construction.



Classified Advertisements

The rate for all classified advertisements is 6d. per word; in bold print 9d. per word; minimum order 6s. Box-number advertisements 1s. extra. Instructions, together with remittance, must be received not later than the 3rd of each month for advertisements to appear in the same month's issue.

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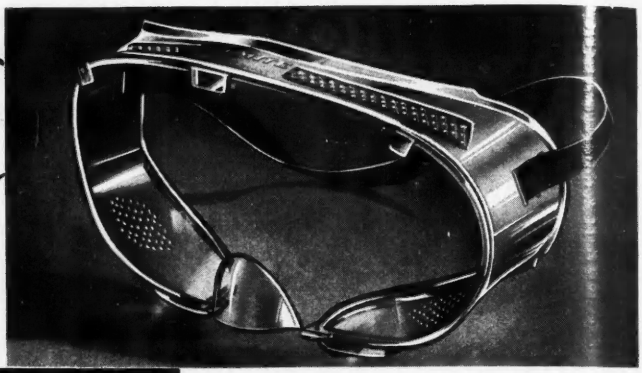

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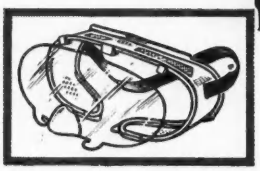

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BOOKS RECEIVED

British Plastics Year Book, 1957. 27th Edition. 716 pages. Publishers: Iliffe and Sons Ltd., Dorset House, Stamford Street, London, S.E.1. Price : 42/- (Postage 1/9).

Fittingly bound in a durable and attractive plastic case, this thoroughly revised 27th edition is, as before, divided into nine sections, three of which are devoted to classified lists of manufacturers and suppliers of materials, finished products, and equipment. Another section comprises the largest list in the world of trade and proprietary names connected with the industry, and covers materials as well as finished products. Each trade name is followed by a definition of the product, the manufacturer concerned, and the country of origin, if overseas. In addition, a glossary of extremely useful technical terms used in plastics is again included in this section.

In the section devoted to names and addresses, details of nearly 4,000 firms associated with plastics are grouped into countries, while the "Who's-Who" section lists the names and positions of prominent people actively engaged in the industry in Great Britain. Another section, of 142 pages, contains reviews of all relevant patents issued last year. These have been conveniently collected into subject groups, each of which is abstracted and prefaced by an expert. In the same section is included a list of new plastics companies registered in 1956, and also British Standard Specifications relating to the industry.

In short, there can be no doubt that the up-to-date information provided in this volume, which remains the only classified guide to products and manufacturers in the plastics industry, makes it invaluable to all connected with the industry and to manufacturers and users of related machinery and equipment.

Electrical Trades Directory, 1957. 75th. Edition. 1605 pages. Publishers : Benn Brothers Limited, Bouverie House, 154 Fleet Street, London, E.C.4. Price : 63/-.

Bearing in mind the enormous volume of material contained in this new edition of The Electrical Journal Blue Book, and the considerable number of new entries incorporated, it is hardly surprising that extremely careful reselection and reconsideration of every entry have had to be made, omitting those which are in any way unconfirmed or which do not comply with the rigorous standards demanded.

The classification and arrangement of sections conform with those established in previous editions, i.e., manufacturing and other companies, consulting, electrical, electronic, heating and ventilating, illuminating, manufacturing electrical, and other categories of engineers, members of the Electrical Contractors' Association, the National Register of Electrical Installation Contractors, the Electrical Wholesalers Federation, the Radio Wholesalers Federation, merchants, importers, exporters, agents, patent agents, electric transport, professional, and trade associations, electricity boards, trade names, products, etc. For convenience, each section, arranged in alphabetical order, has a distinctive heading and is numbered. In addition, an index of all information in the directory, with names arranged in alphabetical order, is included.

The History of Electric Wiring. By John Mellanby, M.A. (Cantab.), A.M.I.E.E., M.Inst.F., M.I.H.V.E. 244 pp., 153 illustrations. Publishers: Macdonald & Co. (Publishers) Ltd., 16 Maddox Street, London, W.1. Price: 32/6.

Although wiring occupies the unique position of being the essential link between supply authorities on the one hand, and designers, manufacturers, and users of lamps and appliances on the other, it has been too often taken for granted. Most developments in wiring were originally undertaken by the manufacturers of cables and other material, who, in the earliest days, were frequently contractors as well, and many accessories were produced in small workshops on the "one-off" principle, as required.

It would appear that no comprehensive history of wiring methods has previously been written. This fascinating book, then, admirably fills the gap, correcting certain prevalent misconceptions, and assigning credit where it is due by not only tracing the invention and development of many devices in common use, but also throwing a little light on the work of a number of personalities who have made notable contributions in this field, including Crompton, Kelvin, Swan, and Edison.

Tracing progress from the year 1870, when electric lighting was virtually confined to the arc lamp, when there were no recognized methods of protecting wiring, when switches, if they existed at all, were crude mechanisms mounted on wood, and when fuses were still to be invented, the author of this delightfully written book not only covers developments in wiring right up to the latest types now installed in prefabricated buildings but also deals thoroughly with accessories and their evolution, including switches, lamp-holders, plugs, sockets, conduits, fuses, etc.

High Speed Diesel Engines. 5th Edition, 1957. By Arthur W. Judge, A.R.C.S., D.I.C., A.M.I.Mech.E., A.M.I.A.E. 578 pp., 429 illustrations. Publishers : Chapman & Hall Ltd., 37 Essex Street, London, W.C.2. Price : 65/-.

Published originally in the early days of high-speed compression-ignition engine development, the different editions of this book which have since appeared have endeavoured to keep pace with the marked and constantly extending progress of this type of engine. As a result, this fifth edition has been largely re-written and extended, so that it contains a considerable amount of new information and a large number of new illustrations.

In the course of this revision, several new chapters have been added, notably those dealing with supercharging, small vehicle engines, air-cooled engines, special-purpose engines, and the starting of engines. As with previous editions, the treatment of the subjects has been kept to elementary standard as far as possible, in view of the purpose of the original appeal to engineers, designers, draughtsmen, and students.

LATEST INDUSTRIAL LITERATURE

1. Iron and Steelworks Equipment. This 20-page illustrated brochure shows the extensive range of ancillary equipment being made for the rapidly expanding iron and steel industries at home and abroad. Blast furnace plant including ore transfer cars, scale cars, slag ladles and carriages, and pig casting machines, is described. Another section deals with steelworks plant including slag ladles and carriages, hot metal ladles and carriages, steel casting ladles, ingot casting cars and a wide variety of special carriages and bogies.

2. Prestretched V-Belting. Much practical information is contained in a small brochure which details the recommended sizes of belting for the various types of pulley drive, shaft speed, horse power, etc., that are met in everyday engineering.

3. Self-Tapping Inserts. Many materials used today are machinable, but not strong enough to sustain loads on directly tapped threads without shearing. This technical, 8-page, brochure describes a range of case-hardened cadmium plated steel self-tapping bushes which will increase the sheer area and thread resistance. Brass inserts are available for some plastic materials and stainless steel types are supplied to special order.

4. Aluminium Alloys. Having 210 pages of technical data, this new catalogue is of great use to all users of aluminium extruded bars, rods and sections, drawn tubes, forgings and stampings. There are 16 pages of design information while each of the vast number of sections, angles, channels, etc., is well illustrated and dimensioned in detail.

5. Electro-Mechanical Unit Heads. Designed initially for the automotive industry these units are of great advantage where batch production is considered, and they enable increased output, economies in tooling and production uniformity to be achieved in carrying out drilling, milling, reaming, tapping, boring, spot-facing and contouring operations. Three sizes are available for long, medium or short stroke operation and they can be used singly, in combination or in the form of transfer machines.

6. Welding Control. Equipment for the electronic control of resistance welding machines is described in a 20-page illustrated catalogue. Developed to provide a wide range of applications they are specially suitable for automotive and similar industries. Full technical specifications are provided on equipment extending from a simple weld timer for pedestal type welding machines and operated by a limit switch, to complete welding controllers for heavy duty work.

7. Thermoplastic Pipes and Fittings. Detailed in a small folder are three grades of piping which are light, strong and corrosion-proof. The type N is ideal for the process industries, oil and marine engineering. Made with pipe sizes from ¼ in. up to 6 in. it can be used in process and effluent lines up to 100°C under pressure. A special grade is also available for temperatures up to 150°C.

8. Mechanical Vibrator. A new, cheap and very simple vibrator for general engineering use is now in production. Available in a range of sizes it consists of a circular diecast box containing a steel ball which is driven round a track by air pressure from the standard air feed line. Using air pressures up to 100 psi a thrust of about 20 lb can be obtained at vibration frequencies up to 100 cycles per second. This unit can be bolted on to such equipment as hopper feeds, conveyor tracks, foundry casting moulds or any machines requiring a medium or small amount of agitation. The need for expensive electro-mechanical vibrators or mechanical couplings from a main driving source is therefore eliminated.

9. Resilient Couplings. This very well produced 32-page illustrated catalogue describes a type of flexible coupling employing a grid spring to connect two discs grooved on the outer edges so that the spring forms a flexible and resilient bridge. The grooves are flared so that the spring members have long flexible spans at normal loads but become supported by the sides of the grooves when overload occurs. Cut-away drawings show the construction in detail while an illustrated section provides a very good description of the methods used to assemble the couplings.

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